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Engineering Problems in Aircraft Operation at High Altitudes

By R. E. JOHNSON[1] and R. F. GAGG,[2] PATERSON, N. J.

The authors review the capabilities of the typical modern airplane for high-altitude operation and present the restrictions of the power plant of the modern airplane which prevent it from delivering its normal output at high altitudes. The paper also discusses the improvements in the aerodynamic characteristics of the airplane as well as changes in the power-plant design and the progress achieved in cruising operation at high altitudes within the limitations imposed by passenger comfort and safety. The authors evaluate further possible gains which may result from an extension of this trend, and examine the engineering problems involved in both the airplane and the engine.

THE striking changes in aircraft performance obtained in the past five years have resulted from material improvements in aerodynamic characteristics, as well as from changes in the power plant. Previously, the latter factor appeared to be nearly the sole means used for improving performance, and the results were rather disappointing, as would be expected from consideration of the simple mathematical relations between speed and power. The quest for increased speed in commercial air transport has turned from a consideration of increased power and improved aerodynamic qualities to a study of the advantages obtainable by extending the cruising operation to higher altitudes. Substantial progress has already been achieved in this manner within the limitations imposed by passenger comfort and safety in present types of equipment, and it now seems in order to evaluate further possible gains which may result from an extension of this trend, and to examine the engineering problems involved in both the airplane and the engine.

The typical modern airplane is capable of normal flight operation at altitudes far higher than any now utilized if a suitable power plant were available which could deliver its normal power output at any altitude. Since the restriction of primary importance is in the power plant, that problem is considered before counting the gains which might be realized without this limitation.

Engine Performance Ratings

Three kinds of ratings are required for engines intended for service in high-altitude aircraft. These are (1) take-off rating, based on the maximum permissible power output for short periods at low altitudes; (2) maximum power for protracted periods, which power is to be used to reach destination in event of failure of one or more power-plant units and not for normal operation for long periods; and (3) cruising rating, which rating requires maximum power for continuous normal operation.

In order to provide a quick and safe take-off, it is desirable that the take-off rating be 150 per cent of the cruising rating. This large margin of reserve power will enable the airplane to operate from small airports, and also provides a factor of safety for emergencies. The intermediate rating, termed the maximum power for protracted periods, is not normally required in airline service, and is used only in case of a continuing need for a maximum of power consistent with safe operation, such as the failure of one unit in a multiengined airplane. This high-power rating obviously is required only at altitudes where some high land may be encountered in the flight path. The cruising rating

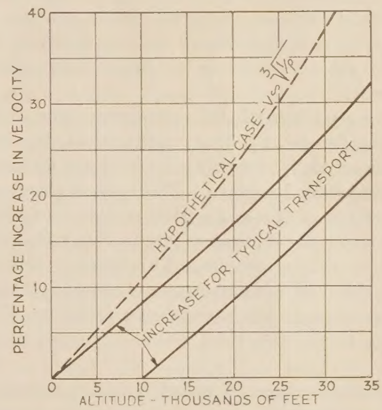

Fig. 1 Change in Velocity of Level Flight With Constant Power for Various Altitudes

and the limiting height at which that power can be maintained are the most important measures of the usefulness of an engine for high-altitude operation. When used in the same kind of airplane, the same speed will be obtained with a 500-hp engine at 12,000 ft altitude or an engine giving 330 hp at 35,000 ft. In terms which have a greater appeal for the American operator, if the 500 hp were maintained to 35,000 ft the speed would increase by about 23 per cent. These relations are illustrated in Fig. 1 for a modern transport airplane. Without considering other important factors, it would, therefore, appear that unlimited upward extension of the altitude limitation of cruising power would be desirable. Notable progress has been achieved in this direction as the related problems of durability and power output, supercharging, combustion control and cooling, and fuel consumption have gradually been disposed in harmonious balance. Fig. 2 shows changes in the ratings of a typical American engine resulting from assiduous demands for greater power output and increased altitude performance with improved durability. A brief examination of current limitations on further progress will indicate what may reasonably be expected from the engine in the

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near future and thus establish the field immediately to be considered for high-altitude flight.

Assuming that the engine design is properly proportioned with respect to strength, heat flow, and lubrication, its cruising rating will be established in practice at the highest level which will permit operation without major failures in a period of 350 to 400 hours between overhauls. In other places it is sometimes the practice to operate at a lower output and stretch the overhaul period, but this is scarcely to be expected in American operations. Types of engines now available are capable of cruising ratings of 0.25 to 0.33 hp per cu in. of displacement, and the upper value is likely to change but slowly for a considerable period unless radical departures in engine design are encountered.

Engine Supercharging

With a supercharger of the centrifugal-compressor type driven from the engine crankshaft at a fixed gear ratio, it is feasible to maintain the cruising power rating up to about 14,000 ft altitude and still obtain acceptable take-off performance with present types of fuel. This range can be somewhat extended with the aid of an intercooler, though the additional drag, weight, and complication do not appear to justify its use. The limitation of the single-stage supercharger just mentioned is encountered at sea level at the take-off condition. The liberal use of fuel cooling and tetraethyl lead in the fuel during take-off has considerably extended the useful range of this type of supercharging, but even with benefit of much improved cooling practice and better fuel it is apparent that the difficulties encountered under take-off conditions will place a relatively low limit on its utility for high-altitude aircraft.

A second geared centrifugal compressor used only at high altitudes to supplement a primary unit of the type just discussed has been used successfully to obtain additional altitude performance with conventional engines. This type of unit has all of the advantages of the single-stage unit at low altitudes plus the ability to maintain the cruising rating to about 25,000 ft altitude without the use of an intercooler. This improvement in performance can be obtained without material increase in the airplane drag, though it does require a somewhat higher specific fuel consumption due to the increased power absorbed by the supercharger.

Many of the advantages of both the dual-compressor unit and the integral, single-stage supercharger can be realized by the incorporation of a two-speed transmission with suitable clutching in the gear drive for the integral-type of centrifugal supercharger. Such an arrangement permits optimum sea-level performance for take-off, and by changing the speed of the supercharger impeller, gives adequate supercharging for high-altitude cruising.

The second or high gear ratio may be chosen so that the supercharger compression ratio is the maximum that can be used with the engine and fuel selected without encountering detonation in cruising. Assuming the same compressor efficiency in each case, the altitude limitations of the two-stage compressor and the two-speed single-stage compressor are, therefore, identical without intercooling, and the latter refinement scarcely seems warranted in the face of the high penalties which its use imposes. For the ultimate maximum in degree of supercharging, two-stage compression with an intercooler may prove to be necessary, but until the means otherwise available have been exhausted this complication should be avoided.

The combination of one supercharger stage gear-driven from the crankshaft, and another stage driven by an exhaust-gas turbine may be compared fairly to the two-stage gear-driven combination. The low-altitude performance ratings may be obtained by the use of the gear-driven supercharger only, exactly as in the cases previously considered, and the turbine unit may be used at high altitudes to maintain the cruising-power output. Due to the fact that the turbine speed, and hence the supercharger compression ratio, is subject to control, the required power may be obtained at intermediate altitudes with a little lower charge temperature than that obtained with the two-stage gear-drive unit.

It seems probable that the combination gear and turbine-driven two-stage supercharger will give excellent performance at very high altitudes, but the complication of the dual-type installation plus the practical problem presented by an exhaust manifold operating at an unusually high temperature and pressure are obstacles which may block its progress. Moreover, there is no conclusive evidence to prove that the results obtained without intercooling cannot be matched by a two-stage geared supercharger. These handicaps appear to preclude its wide adoption in the immediate future.

The exclusive use of an exhaust-turbine drive for the supercharger at all altitudes does not appear to be particularly attractive because of the difficulty of obtaining a very high output for take-off at sea level without encountering prohibitive temperature and pressure in the exhaust system and cylinders.

It is unfortunate that necessary experimental data are not available as a basis for completely evaluating the merits of the turbo-supercharger. However, it seems probable that this type, when used with adequate intercooling, will produce the highest power output at altitudes above 30,000 ft.

For the near future, it therefore appears to be feasible to employ either two-stage compression or a two-speed supercharger drive which will permit attainment of adequate power ratings for take-off and emergency service at low altitudes with the lower of the two degrees of supercharging, and to maintain the cruising-power rating only to about 25,000 ft altitude with the other combination. Such an engine requires the use of a propeller that is completely controllable and a governing mechanism to maintain the engine-operating speed at suitable values. With this arrangement the maximum airplane-cruising speed will be obtained at 25,000 ft and the speed at 35,000 ft altitude will be about 94 per cent of the maximum. This range would normally be used for cruising operation, and if circumstances require low-altitude running, the engine will be well adapted for operation at the same cruising rating near sea level.

Consideration of the numerous variables which need to be controlled under the wide range of operating conditions indicates that unless intercooling is necessary to obtain the required output, the optimum of performance can only be realized through use of a variable-compression supercharger with complete control of its operation in flight.

Satisfactory Cooling Attained With Supercharging

An improved comprehension of the means for controlling combustion and cooling in aircraft-engine practice has effected an increase in the performance attainable with a given fuel both at sea level and higher altitudes, as is shown by the curves of Fig. 2 for the period 1930 to 1935. Modern engines generally operate at temperatures equal to or lower than those commonly experienced in 1930 in spite of higher specific power output. It is interesting to note that expected difficulties with the cooling of supercharged engines in general have not been realized in practice, except in cases where abnormally low air speeds were encountered, as in high power output at very low airplane speeds. Normal air-line operation fortunately does not require this low-velocity climbing, and, as previously mentioned, the cruising-power rating is the maximum required at high altitudes.

Fuel Consumption a Critical Problem

Since high-altitude flight is obviously not particularly advantageous for short distances between stops, the total propulsion weight is a matter of vital importance. This weight comprises that of the propeller, engine, cooling apparatus, installation, fuel, and oil. All items except fuel and oil in the tabulation are, in general, determined by necessities of the required power output and durability. The fuel consumption at the cruising-power rating is the dominant variable in the propulsion-weight figure for a flight of more than four or five hours. Actual realization of specific fuel-consumption values of 0.42 to 0.45 lb per bhp-hr under normal cruising conditions is practicable with present engines and fuels where combustion control, heat dissipation, and operating technique are properly coordinated. These figures are obtainable with an overall mechanical efficiency of about 88 per cent. With the same fuel-air ratio, making appropriate correction for the increase in power input to the supercharger, the same engine when supercharged to give the cruising rating at 25,000 ft altitude will give a minimum specific fuel consumption of 0.48 to 0.51 lb per bhp-hr. This severe penalty in higher minimum specific fuel consumption to attain good performance at high altitudes greatly restricts the field of usefulness of this type of engine and will result in a compromise in the selection of the operating altitude for a contemplated route schedule. However, even the values just quoted are not higher than the average experience of air-line operators with present methods of control. In order to realize the minimum of fuel consumption it is necessary to install precise control apparatus and to exercise meticulous supervision. Even though the fuel consumption while cruising warrants the most precise control, to use lean-mixture ratios for take-off and other high-power operation is fallacious. The richest mixture which can be employed without loss of power should be used for the take-off in order to provide a maximum factor of safety against unforeseen contingencies. In any case, the amount of fuel wasted by this procedure is insignificant for a long flight.

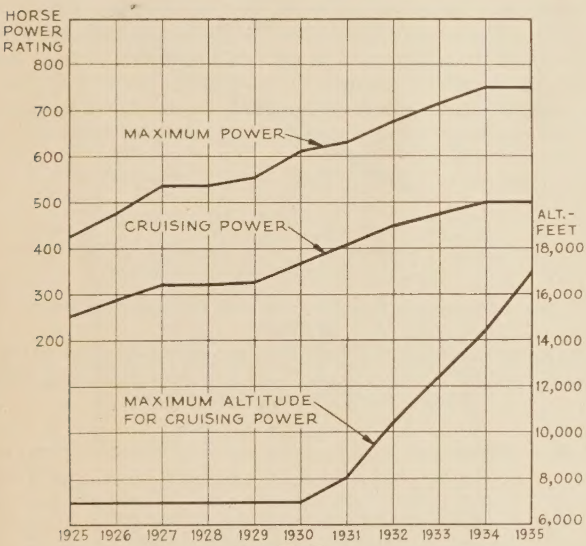

Fig. 2 Changes in Rating of a Typical Airplane Engine

If fuels of improved knock rating become generally available, it will be possible to increase the compression ratios now generally used and thus improve both the power output and the specific fuel consumption. This practice will almost certainly introduce a whole new series of operating durability problems. However, a decrease of 10 per cent in minimum attainable fuel consumption for cruising is worth a major effort because of the resulting large increase in pay load or range.

There are undoubtedly many operating difficulties now unforeseen which will be encountered when aircraft engines begin routine service at high altitudes. The ignition system and the fuel-supply problem may prove troublesome, but it is unlikely that these, or similar difficulties will be insurmountable. Power plants suitable for cruising operation at 25,000 ft to 35,000 ft altitude appear to be feasible with fuel of 87 octane number and a conventional type of engine within a period of two or three years, but the specific fuel consumption will be relatively high because of the power consumed by the supercharger.

Time Saved by High-Altitude Cruising

Having thus broadly outlined the limitations established by the power plant for high-altitude aircraft of the near future, an estimate may be made of what changes in useful performance may be realized within these boundaries. It will be noted that the feasible modifications in the power plant previously mentioned do not involve any direct increase in drag, but only an increase in size and weight of auxiliary equipment, which probably can be accommodated within the space available in an air-line transport airplane. It is reasonable, therefore, to compute the changes in performance from available data on the assumption that the basic flight characteristics of a transport airplane of modern type will not be altered materially by the installation of an advanced type of supercharged engine.

The increase in cruising speed of a transport airplane equipped with engines which will maintain constant cruising-power output to 25,000 ft is shown in Fig. 1. It will be noted that this increase in level cruising speed at 25,000 ft over that of the same airplane operating at 10,000 ft is 13 per cent. This increase will not vary greatly for any modern type of airplane with conventional wing loading. In determining the practical utility of the high-altitude airplane, the apparent increase in performance resulting from high-altitude cruising must be tempered by consideration of the trip length and schedule speed, rather than level cruising speed alone.

In order to obtain the reduction in station-to-station time possible by cruising at altitudes higher than 10,000 ft, it is necessary to have certain basic performance data for the airplane, similar to that shown in Fig. 3, for the engine-operating technique normally employed. For the purpose of this analysis, the following assumptions have been made to obtain the comparative station-to-station time or speed:

(1) A five-minute interval has been allowed for maneuvering the airplane at each end of the trip. This total of ten minutes maneuvering time has been found by experience with modern transport operation to be a necessary part of the schedule for any flight.

(2) All climbs are made with a constant-thrust horsepower which is 10 per cent higher than the normal cruising power. It has been assumed that the engine will be capable of supplying this power to at least 25,000 ft altitude. The rate of climb is determined in the usual manner from the power required and power available curves for the airplane being considered. It has also been assumed that all climbs will be made at constant indicated air speed.

(3) All level cruising will be done at the established cruising-power rating for the engine.

(4) All glides will be made at constant rate of descent, using the rated cruising power.

Fig. 3 shows performance data for a typical modern transport airplane operating as outlined above. From these data, the

station-to-station time or speed may be calculated for any trip length and for any assumed cruising altitude. Fig. 4 shows the results of these calculations for 10,000, 25,000, and 35,000 ft cruising altitudes. Inspection of these curves will give the time saving in minutes which may be expected for any trip length for the airplane which cruises at 25,000 ft compared to the same airplane

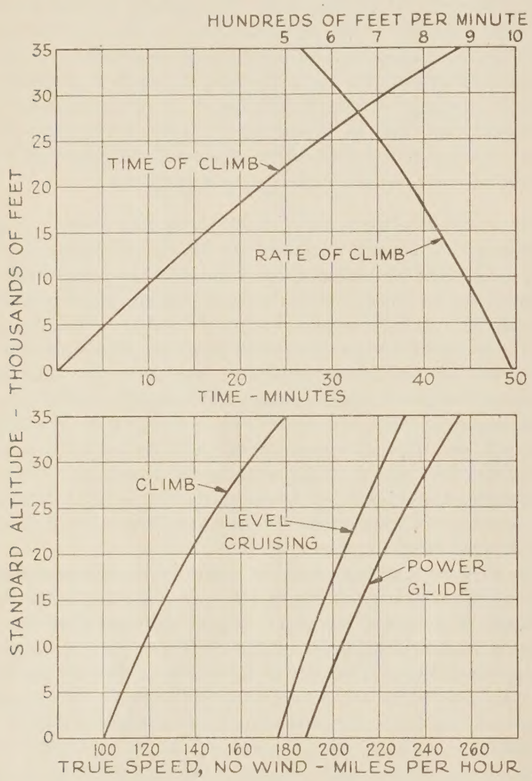

Fig. 3 Cruising-Performance Data of a Typical Transport Airplane

cruising at 10,000 ft with the same horsepower. This represents the real advantage in schedule-time reduction available to the airline operator.

Fig. 5 shows this time saving in minutes for any trip length for the particular airplane being considered, and also the percentage of schedule-time reduction possible for the 25,000 ft cruising over the 10,000 ft operation which may be realized approximately in any airplane operating under similar conditions.

If these data are interpreted in terms of a typical transcontinental air-line schedule, it may be expected that this airplane, cruising at 25,000 ft will be able to make the trip without stopping in about 80 min (9.5 per cent) less time than the same airplane flying at 10,000 ft with the same power. If the airplane stops once during the transcontinental flight, it may be expected to save 74 min (9 per cent) by cruising at the higher altitude. If it stops three times, it may be expected to save 52 min (6 per cent). If stops are made once every 300 miles, there is practically no advantage in going to 25,000 ft for cruising.

This indicates that the practical advantage in time saved by cruising at a high altitude is not large except for very long flights such as are necessary in transoceanic service. In the latter operations, the time saved is not particularly important except in regard to its effect on the amount of fuel which must be carried. For any specific flight length, the fuel-load reduction credited to the time saved must be corrected for the higher specific fuel consumption necessitated by supercharging for the higher-altitude operation, and for other weight penalties mentioned later. For operation over land, the time saved in making long flights at high altitudes is not ordinarily of as great importance as the advantages of increased pay load obtainable by making an occasional stop for fuel.

If restrictions on the rate of descent imposed by passenger comfort and safety are eliminated (as by use of a supercharged cabin), a slight additional time saving may be accomplished by cruising at high altitudes, but this does not materially alter the data as presented.

Pressure Cabin Required for Passengers

Since experience has indicated that passengers cannot ordinarily be carried at altitudes higher than 12,000 or 14,000 ft without increasing the air pressure in the cabin above that of the

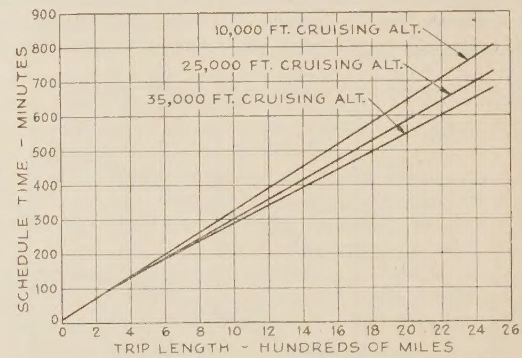

Fig. 4 Schedule Performance of a Typical Transport Airplane

(The station-to-station time includes (1) 10 min maneuvering time, (2) time to climb to cruising altitude with 110 per cent cruising power, (3) time at cruising altitude with cruising-power rating, and (4) time gliding from cruising altitude with cruising-power rating.)

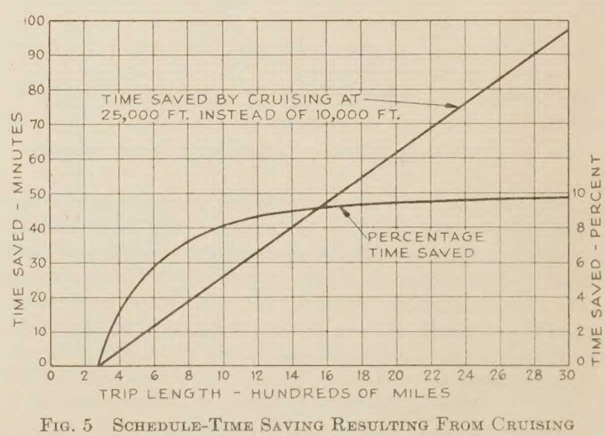

Fig. 5 Schedule-Time Saving Resulting From Cruising at High Altitude

surrounding atmosphere, the fuselage must be substantially airtight for high-altitude cruising. This is not impractical, though it involves some increase in structural weight. In order to minimize difficulties, it would seem advisable to maintain the cabin pressure at about 10 lb per sq in. abs at altitudes above 10,000 ft, at which altitude that atmospheric pressure is the normal value. If this procedure is followed, the internal pressure which the cabin must withstand will be nearly 7 lb per sq in. abs at 35,000 ft altitude. The stiffness required of the normal fuselage is such that the conventional structure will, in general, be adequate for

the bursting loads. The principal structural changes needed will occur at bulkheads and doors.

A troublesome problem is encountered in arrangements for maintaining the cabin pressure and in providing for ventilation. The power needed by a centrifugal compressor to raise the pressure from atmospheric to 10 lb per sq in. abs is shown in Fig. 6, for the range of flow ordinarily considered acceptable in ventilation practice. Assuming that a value of about 20 cu ft per min per passenger is satisfactory, the power required will be about 2 hp per passenger at 35,000 ft. The utilization of the main power plant to supply this energy is immediately suggested for economy of both structural weight and fuel. In the case where a two-stage supercharger is used for the engine, the first stage might possibly supply the cabin pressure and ventilation with the assistance of pressure- and volume-control apparatus. A small auxiliary compressor driven by an exhaust-gas turbine would be both economical of fuel and flexible in operation.

In addition to the desired cabin pressure and ventilation, a large portion of the heat required for the comfort of the passengers would be supplied by the same centrifugal compressor. While this amount of heat would have to be supplemented by an exhaust-heated steam boiler or other appropriate means, the heated-air supply would furnish a large part of the requirement except at low altitudes. Under unusual atmospheric conditions, cooling of the cabin air supply might even prove necessary as indicated by the temperature data of Fig. 7. The high-temperature air must be carefully distributed in the cabin to avoid uncomfortable hot spots.

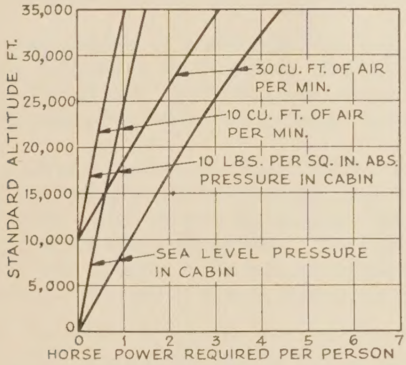

Fig. 6 Horsepower Required per Passenger for Supercharging Cabin

Automatic devices are required to control the cabin pressure and temperature, and a spare cabin supercharger may be necessary to protect the safety of passengers in case of a mechanical failure. If it is desired to employ high rates of descent to save time, it will be necessary to insure a gradual change of pressure within the cabin. This may be accomplished by temporarily raising the absolute pressure in the cabin above 10 lb per sq in. abs just prior to landing, or by decreasing the rate of descent at low altitudes. The continual use of sea-level pressure in the cabin involves both additional structural-weight changes and an increased power consumption. The number of devices required for cabin supercharging and pressure control again emphasizes that an airplane intended for cruising at high altitudes must be a relatively large unit in order to achieve a reasonable pay-load capacity.

Difficulty with boiling of the fuel at low pressure will be aggravated at higher altitudes and it will probably be necessary to apply cabin pressure to the fuel system. This matter and numerous similar detail problems will need careful attention if successful operation is to be achieved. The precision of navigation instruments, for example, is a matter of increasing importance as speeds and distances are increased. However, it is believed that the solution of any of these problems does not present major difficulties.

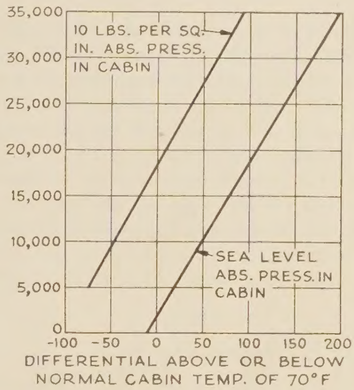

Fig. 7 Temperature of Air Discharged From Cabin Supercharger

Conclusion

The preceding discussion leads to the conclusion that the additional complications in structure and auxiliary apparatus required for high-altitude cruising with passengers probably does not warrant that type of flight operation except in the case of long non-stop flights over water, or to avoid or make use of weather conditions on the flight route. However, the latter item may be a very important consideration when it is desired to take advantage of prevailing winds. If these special considerations induce the construction of a highly specialized, pressure-cabin type of airplane for high-altitude operation, the flight path normally used (neglecting winds) should be at the maximum altitude permitted by the degree of supercharging of the power plant. It seems probable that the supercharging methods available for service use in the near future will not provide for the maintenance of the cruising-power rating at altitudes greater than about 25,000 ft, and that the fuel consumption of such an engine will be materially higher than that of a similar engine supercharged for operation at relatively low altitudes. This increased fuel consumption plus the additional penalties resulting from the necessity for a supercharged cabin, severely restricts the commercial utilization of the increase in airplane speed achieved by flight at high altitudes.

The Flow Characteristics of Variable-Speed Reaction Steam Turbines

By ADOLF EGLI,[1] PHILADELPHIA, PA.

In this paper a general relation is derived between weight of flow, speed, and efficiency of a reaction-blading group operating under various steam conditions. With this relation, which is best plotted graphically, the author presents a method for calculating the peculiar variation of the weight of flow in variable-speed turbines.

IT IS a well-known fact that the weight of steam flowing through a reaction turbine varies peculiarly with the speed of the turbine. This variation is appreciable, particularly in turbines used for marine propulsion which operate over an extremely wide range of speeds. When predicting the performance of such turbines[2] the effect of the speed on the pressure distribution in the turbine, i.e., the effect on the weight of flow through each group of blades, should indeed not be neglected. Due to the extremely complicated nature of the problem, however, this refinement in turbine calculations has been neglected generally.

The principal solution of the problem has been given by Stodola[3] and is known as the v^2-method. This method is applicable to any blade path. It involves, however, a cumbersome cut-and-trial calculation which must be made for each row of blades.

A general relation between the characteristic variables of a whole reaction-blading group with uniform blade profile and uniform velocity ratio is derived in this paper which for all practical purposes is sufficiently accurate. The type of blading mentioned is, with exception of the last few rows in condensing turbines, the only one used in modern reaction turbines.

The Flow Through a Symmetric Reaction-Blading Group

A symmetric reaction-blading group is made up of blades with the same profile in all blade rows, i.e., blades of geometrically similar profiles, and is designed so that when operating with the design pressure ratio, all velocity triangles are geometrically similar.

1 *The Velocity Ratio ν as Characteristic Argument of the Performance of the Blade Group.* Consider two-dimensional flow through one row of rotating or stationary blades as seen by an observer standing still relative to the blades. The flow picture, which looks principally as sketched in Fig. 2, is, in accordance with the laws of similarity of flow, determined if we give the angle of approach β, the Reynolds number R,[4] and the Mach number M.[5] For one and the same turbine the effect of R is very small, unless it is operating in a critical flow region, which generally is not the case. In the analysis presented in this paper the effect of the Reynolds number on the performance of

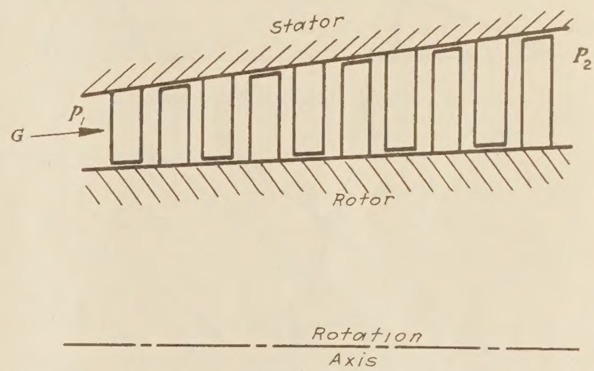

Fig. 1 Reaction-Blading Group of an Axial-Flow Turbine

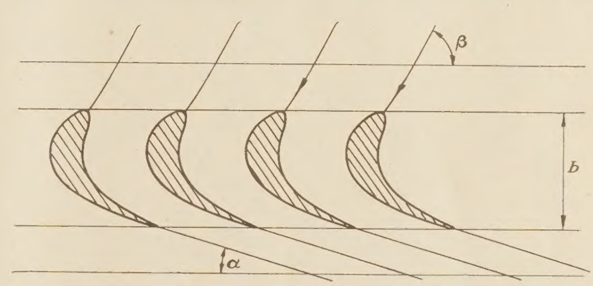

Fig. 2 Two-Dimensional Flow Through a Row of Reaction Blades

the blading is neglected. For such blade groups, the Mach number is usually about 0.3 or less so that its influence on the efficiency can also be neglected. The steam, when flowing through the blade row, behaves practically as an incompressible fluid.

Thus, the picture of the flow through the blade rows of a given turbine is assumed to be dependent only on the angle of approach β. In particular, consider now the approach velocity c_2 and the leaving velocity c_1. The direction of the two velocities and the ratio of their absolute values are fixed as soon as β is

[1] Research Engineer, Westinghouse Electric and Manufacturing Company. Mr. Egli received his technical education at the Federal Technical University in Zürich, Switzerland, where he was graduated as Diploma Mechanical Engineer under Professor Stodola in 1929. After graduation, he was engaged as assistant to Professor Stodola and Professor Eichelberg and as instructor of thermodynamics and heat-engine design. From 1930 to 1931 he was employed as experimental engineer with the Terry Steam Turbine Company, Hartford, Conn. Since 1931, Mr. Egli has been connected with the Westinghouse Company at South Philadelphia as research engineer on turbine efficiency and design.

[2] Steam-rate guarantees are often required over a range of loads from the maximum down to less than 2 per cent with the heaviest penalties at the lowest loads.

[3] "Steam and Gas Turbines," by A. Stodola, McGraw-Hill Book Company, New York, N. Y., 1927, pp. 316–327.

Contributed by the Power Division and presented at the Annual Meeting of The American Society of Mechanical Engineers held in New York, N. Y., December 2 to 6, 1935.

Discussion of this paper should be addressed to the Secretary A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1936, for publication at a later date.

[4] Defined as $R = cl/\nu$, where c = velocity at a given point of the flow field, l = a characteristic length such as the blade width b, and ν = the kinematic viscosity of the fluid.

[5] Defined as $M = c/c_a$, where c and c_a are the flow velocity and acoustic velocity, respectively, at a given point in the field.

given. Neglecting the difference between the mean diameters of two consecutive blade rows, the relation between the velocities c_1 and c_2 and their directions α and β is represented by the velocity triangle shown in Fig. 3. In this figure u represents the relative blade speed between two consecutive rows, i.e., in an axial-flow turbine with rotor and stator u is the peripheral velocity of the rotating blade. It is apparent that the proportions of the velocity triangle are definite functions of the approach angle β, and also the velocity ratio $\nu = u/c_1$. Besides Reynolds' number and other factors, taking into consideration the effect of varying blade heights, blade widths, etc., it is common practice to use the velocity ratio instead of the approach angle as characteristic argument of the performance of a turbine stage.

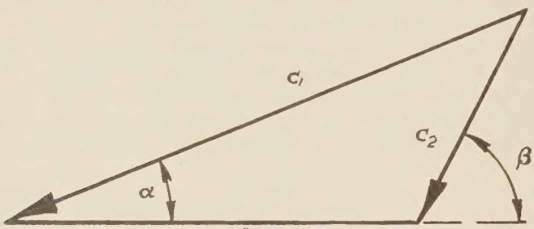

Fig. 3 Velocity Triangle

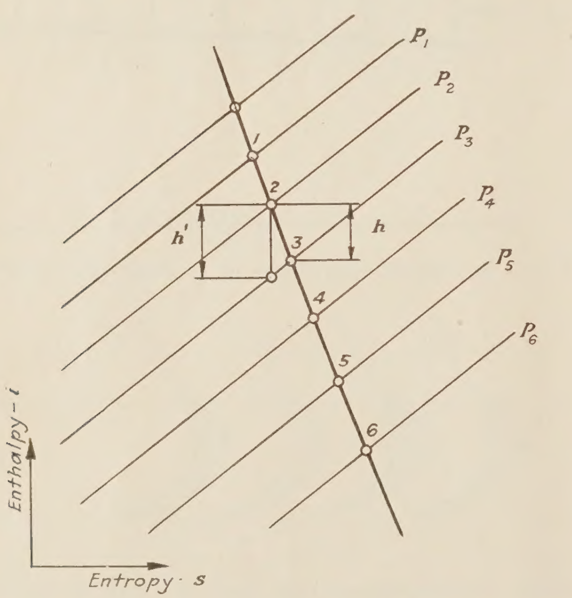

Fig. 4 Condition Curve for the Flow Through the Blade Group

For the special type of blade group considered in this paper the author will follow the common practice and use ν instead of β as characteristic argument even though the approach angle β is a more general characteristic. The variation of the factors for blade height, blade width, etc., do not have to be included since we are concerned only with a given blade group. The flow through the blade row is now a function of the velocity ratio, i.e., each velocity vector is a function of ν; the percentage energy lost through friction is a function of ν; the efficiency of the flow is a function of ν; the angle α which the flow assumes after it has left the blade row is a function of ν; etc.

A symmetric blading group is designed so that each row is operating with the same velocity ratio, i.e., all velocity triangles are geometrically similar.

2 *The Variation of the Blading Efficiency of a Given Blade Group.* The points 1, 2, 3, 4, 5, 6, etc., in the enthalpy-entropy diagram of Fig. 4 represent the condition of the steam at the exit of the first, second, third, fourth, fifth, sixth, etc., blade row. Then the blading efficiency of the blade row is defined as

$$\eta = h/h' \qquad [1]$$

where h is related with the velocities of the triangle of Fig. 4 through the energy equation

$$h = (A/2g)(c_1^2 - c_2^2) \qquad [2]$$

where $A = 1/777.6$ Btu per ft-lb, and $g = 32.17$ ft per sec per sec.

To express numerically the energy losses in the flow through the blade row we introduce with Stodola the velocity coefficient φ. The coefficient φ is the ratio of the actual leaving velocity c_1 to the velocity c_1' obtained in a flow without energy losses. It is defined by the equation

$$(A/2g)c_1^2 = \varphi^2[h' + (A/2g)c_2^2] \qquad [3]$$

The velocity coefficient φ is definitely related to the energy losses n the flow, i.e., φ is a function of ν.

Substituting h from Equation [2] and h' from Equation [3] in Equation [1]

$$\eta = \frac{1 - (c_2^2/c_1^2)}{(1/\varphi^2) - (c_2^2/c_1^2)} \qquad [1a]$$

Noting the relation from the geometry of the velocity triangle

$$c_2^2/c_1^2 = 1 - \epsilon \qquad [4]$$

where $\epsilon = \nu\ (2 \cos \alpha - \nu)$, a function of ν, we write Stodola's expression for the blading efficiency

$$\eta = \frac{\epsilon}{\epsilon + (1 - \varphi^2)(1/\varphi^2)} \qquad [1b]$$

For a given blading, the efficiency η is a function of the velocity ratio ν only, since ϵ as well as φ are definite functions of ν.

The relative kinetic energy available for the flow through the following blade row will be calculated at this point since it is needed later in the derivation of the flow characteristics. It is defined as

$$e = (A/2g)(c_2^2/h')$$

and can be expressed in terms of φ and ϵ in the form

$$e = \frac{(1 - \epsilon)\varphi^2}{1 - (1 - \epsilon)\varphi^2} \qquad [5]$$

where e is also a function of the velocity ratio ν.

3 *The Isentropic Velocity Ratio* ν'. When calculating the turbine performance, it is not possible to compute directly $\nu = u/c_1$ since c_1 is unknown. In practical calculations one therefore refers to the adiabatic velocity ratio

$$\nu' = u/c_0$$

where

$$c_0 = \sqrt{(2gh'/A)} \qquad [6]$$

The adiabatic velocity ratio ν' can easily be calculated since both the enthalpy drop h' (at constant entropy) across the stage and the turbine speed are known. The relation between ν' and ν is obtained by writing Equation [3] in the form

$$1/\varphi^2 = (c_0/c_1)^2 + (c_2/c_1)^2 \qquad [7]$$

and noting that

$$c_0/c_1 = \nu/\nu'$$

Thus

$$\nu/\nu' = \sqrt{[\epsilon + (1 - \varphi^2)(1/\varphi^2)]} \dots\dots\dots\dots [8]$$

Combining Equations [1b] and [8] as well as [4] and [8], yields the general relations between the variables (ν/ν'), e, φ, η, and ν' for instance in the form

$$\nu/\nu' = \text{function of } \eta \text{ and } \nu' \dots\dots\dots\dots [a]$$

$$e = \text{function of } \eta \text{ and } \nu' \dots\dots\dots\dots [b]$$

$$\varphi = \text{function of } \eta \text{ and } \nu' \dots\dots\dots\dots [c]$$

where η and ν' are most readily used as parameters since η is directly obtainable as a function of ν' from turbine tests. The actual numerical evaluation of the three functions [a], [b], and [c] is best performed graphically. In order to demonstrate their principal features the author has calculated them for a fictitious blading for which $\sin \alpha = 0.30$ shall be a constant, independent of the velocity ratio ν. The constancy of α with ν over a wide range of operating conditions approximately is the case in actual blading.

However, it is not very difficult to include in the calculation of these characteristic functions any variation of α with ν, which for instance may have been found by special tests.

In Fig. 5, the ratio (ν/ν') has been plotted as a function of η and ν' for blading with the $\sin \alpha = 0.30$. This particular curve is useful in the design of the blade path, ν' being known from the performance calculations and ν being directly necessary when calculating the blade heights. Figs. 6 and 7 show the percentage of carried-over kinetic energy e and the velocity coefficient φ, as functions of η and ν', respectively.

It should be noted that Figs. 5, 6, and 7 are general and apply to any blading group made up of blades which produce an exit-flow angle α for which $\sin \alpha = 0.30$.

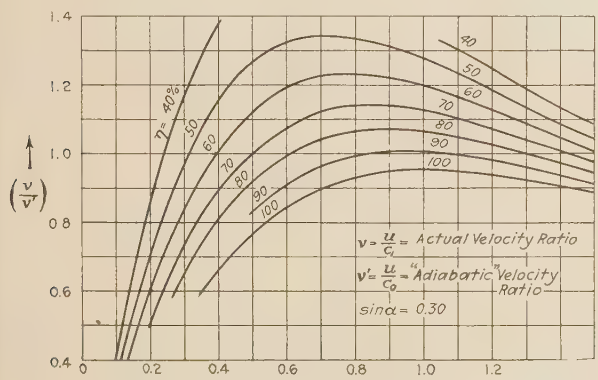

Fig. 5 Actual and Adiabatic Velocity Ratio for Blading With $\sin \alpha = 0.30$

4 *The General Continuity Equation.* This equation shows the relation between weight of flow, speed, blading efficiency, and pressure distribution of a group of reaction blades operating under various steam conditions.

There are two principal ways in which the flow through the blade row appears to take place. Over a certain range of approach angles the flow picture looks about as shown in Fig. 2, the fluid following the contour of the blades without loss of contact. Below a certain value β_a of the approach angle and above a certain value β_b, the boundary layer breaks loose from the blade

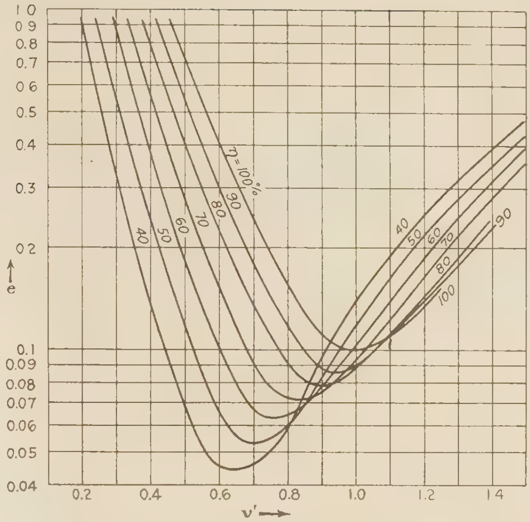

Fig. 6 Percentage of Kinetic Energy e Available From Preceding Row for a Blading With $\sin \alpha = 0.30$

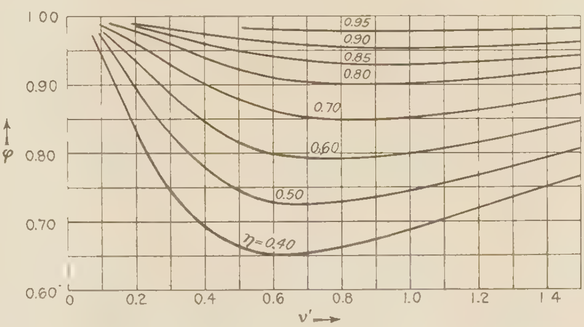

Fig. 7 Velocity Coefficient φ for a Blading With $\sin \alpha = 0.30$

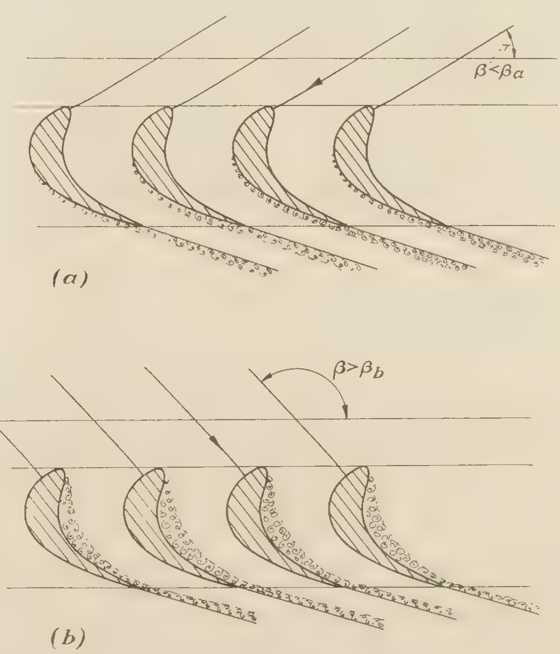

Fig. 8 Flow Through a Row of Reaction Blades With Loss of Contact

wall and a wake of eddies is produced after each blade as shown in Figs. 8a and 8b. In the continuity equation

$$Gv = fFc_1 \qquad [9]$$

where G = weight of flow, lb per sec; v = specific volume of the fluid at the exit from the blade row, cu ft per lb; F = flow area at the exit from the blade row, sq ft; c_1 = exit velocity from the blade row, ft per sec; and f = a fill factor which takes into account the extent to which the flow passage is filled. In the range of approach angles β_a to β_b (corresponding to velocity ratios ν_a to ν_b), f will be unity. In the regions below ν_a and above ν_b, f will be less than unity. We should expect f to vary with the velocity ratio about as shown in Fig. 9.

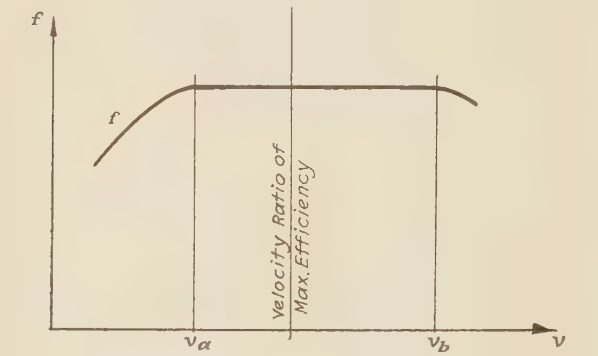

FIG. 9 FILL FACTOR f AS A FUNCTION OF THE VELOCITY RATIO

By eliminating c_1 from the Equations [3] and [9] and introducing e as defined with Equation [5]

$$G^2 = (2g/A)F^2f^2\varphi^2(1 + e)(h'/v^2) \qquad [10]$$

Inasmuch as h', the isentropic enthalpy change across one blade row, is relatively small we can write approximately

$$h' \cong -\Delta i = -Av\Delta p \qquad [11]$$

where Δp is the pressure drop across the row. Equation [10] can now be written in the form

$$G^2/2gf^2\varphi^2(1 + e) = -F^2(\Delta p/v) \qquad [10a]$$

in which it is observed that the left side of Equation [10a] has for each row of a symmetric blading group the same value

$$D = G^2/2gf^2\varphi^2(1 + e) \qquad [12]$$

The equation

$$D = -F^2(\Delta p/v)$$

may also be written in the form

$$D\frac{1}{\Delta x}\frac{1}{F^2} = -\frac{1}{v}\frac{\Delta p}{\Delta x} \qquad [12a]$$

with Δx as an increment of the number of rows. We then introduce but a small error when replacing $\Delta p/\Delta x$ by dp/dx so that

$$D\frac{1}{\Delta x}\frac{dx}{F^2} = -\frac{dp}{v} \qquad [13]$$

Integrating Equation [13] over the whole group of blades which is operating at a fixed velocity ratio

$$D\frac{1}{\Delta x}\int\frac{dx}{F^2} = -\int_{p_1}^{p_2}\frac{dp}{v} \qquad [13a]$$

The left side of Equation [13a] becomes a certain fixed value for the particular blade group. The right side has to be integrated along the condition line 1, 2, 3, 4, 5, 6, etc., of Fig. 4. As an approximate relation between p and v along the condition line we may use the polytropic[6]

$$p^nv = p_1^nv_1 \qquad [14]$$

with

$$n = 1 - \eta(k - 1)/k$$

where k = the exponent of isentropic expansion, and η = the blading efficiency. Therefore

$$-\int_{p_1}^{p_2}\frac{dp}{v} = -\frac{1}{p_1^{n}v_1}\int_{p_1}^{p_2}p^ndp = \frac{1}{n+1}\frac{p_1}{v_1}\left[1-\left(\frac{p_2}{p_1}\right)^{n+1}\right] \qquad [15]$$

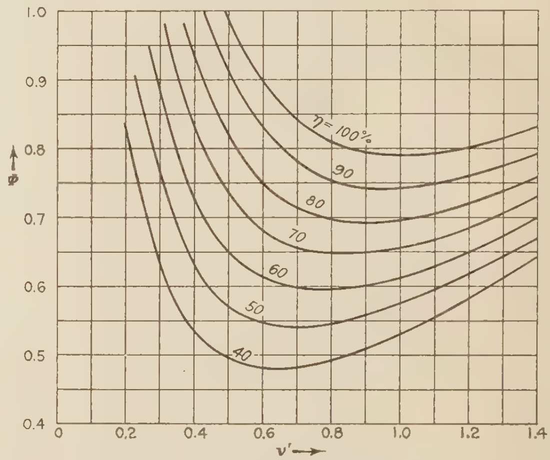

FIG. 10 CHARACTERISTIC FUNCTION $\Phi(\nu';\eta)$ FOR A BLADING WITH SIN α = 0.30

and, as result of the approximate integration over the whole symmetric blade group

$$G^2 = \frac{2gf^2\varphi^2(1 + e)}{C^2[2-\eta(k-1)/k]}\left(\frac{p_1}{v_1}\right)\left[1-\left(\frac{p_2}{p_1}\right)^{2-\eta(k-1)/k}\right] \qquad [16]$$

With the simplification

$$2 - \eta(k-1)/k \cong 2$$

the general equation for the weight of steam flowing through a group of symmetric reaction blading is

$$G = Bp_1\sqrt{(g/p_1v_1)}\, f_{(\nu';\eta)}\, \Phi_{(\nu';\eta)}\, \Psi_{(p_2/p_1)} \qquad [17]$$

where B = a characteristic constant for each particular blade group, sq ft; p_1 = absolute pressure before the blade group, lb per sq ft; p_2 = absolute pressure after the blade group, lb per sq ft; v_1 = specific volume before the blade group, cu ft per lb; g = terrestrial gravity, 32.17 ft per sec per sec

$$\Phi_{(\nu';\eta)} = \varphi\sqrt{\frac{1+e}{2-\eta(k-1)/k}}$$

a function of ν' and η

$$\Psi_{(p_2/p_1)} = \sqrt{1-(p_2/p_1)^2}$$

a function of the group pressure ratio plotted in Fig. 11.

[6] "Steam and Gas Turbines," by A. Stodola, McGraw-Hill Book Co., New York, N. Y., 1927 (H. Martin), p. 298; "Festschrift, Professor Dr. A. Stodola," Verlag Orell Füssli, Zürich and Leipzig, 1929 (G. Flügel), p. 145.

For a blading with $\sin \alpha = 0.30$, Φ has been calculated and is shown graphically in Fig. 10 as a function of ν' and η.

The fill-factor f is a function of ν (or also of ν' and η through the relation represented in Fig. 5) depending considerably on the profile of the particular blades used and can be determined directly from efficiency tests with a blading group by comparing G from Equation [17] with the weight of flow obtained from tests. After $f_{(\nu';\,\eta)}$ is known for a particular blade profile, the curves Φ and f can be combined in one $(f\Phi)_{(\nu';\,\eta)}$.

For various given blade groups, η is a known function of ν' and

$$G = Bp_1 \sqrt{g/p_1v_1}\,(f\Phi)_{\nu'}\,\Psi_{(p_2/p_1)} \ldots\ldots\ldots\ldots [17a]$$

The curves $(f\Phi)$ for the variously efficient blade groups will look similar to those shown in Fig. 12. With the use of such curves it is relatively easy to consider the effect of the speed, or better

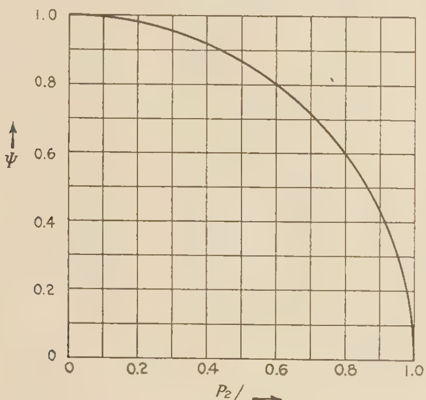

FIG. 11 THE STEAM ELLIPSE $\Psi(p_2/p_1)$

still the velocity ratio, on the weight of flow in predicting the turbine performance.

Special Case. In constant-speed turbines, the variation of the velocity ratio is relatively small during change of load. The blading is generally designed to operate in the flat range 1–2 of Fig. 12 so that the effect of the function $(f\Phi)$ on the flow can practically be neglected. Neglecting, furthermore, the variation of the product p_1v_1

$$G = \text{constant } p_1 \sqrt{1 - (p_2/p_1)^2} \ldots\ldots\ldots\ldots [18]$$

Equation [18] is identical with the "law of the steam cone" found by Stodola.

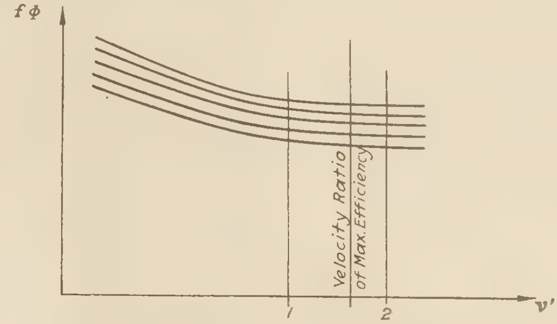

FIG. 12 CHARACTERISTIC FUNCTION $f\Phi$ OF VARIOUS BLADE GROUPS MADE UP OF BLADES OF A GIVEN PROFILE

In case of a condensing turbine, p_2/p_1 becomes negligibly small so that the steam flow G is practically proportional to the inlet pressure p_1.

SUMMARY

The weight of flow through a symmetric reaction-blading group is expressed by Equation [17] to be in general a function of the pressure ratio, velocity ratio, blading efficiency, and initial condition of the steam. Strictly, the integration of Equation [10a] is correct only if the velocity ratio ν is the same for all rows, which is the case if the group pressure ratio p_2/p_1 is equal to the design pressure ratio. For practical purposes, however, Equation [17] may also be applied to varying group pressure ratios, this being especially true since this variation is relatively small as a rule.

Because of the complicated nature of the transcendental functions involved, the characteristic function Φ is best shown graphically. For a given blade section a plot as shown in Fig. 12 will be particularly easy to use in performance calculations.

The curves shown in Figs. 5, 6, 7, and 10 refer, for simplicity, to a blading giving a constant-flow leaving angle α ($\sin \alpha = 0.30$). Principally, any known variation of α with the velocity ratio ν may be considered when calculating the characteristic functions.

When compared with actual tests, Equation [17] has proved to give very accurate results.

Distribution of Air to Underfeed Stokers

By A. S. GRISWOLD[1] and H. E. MACOMBER,[2] DETROIT, MICH.

The development of the underfeed stoker in coal-burning capacity has with few exceptions kept pace with the growth of large boiler units. The ability to continue further the increase in the amount of coal which can be successfully burned will depend greatly upon the solution of the problem of air distribution.

Because of the physical movement and progress of the fuel on an underfeed stoker, and the changes which take place during combustion, the fuel bed presents a progressively varying resistance to air flow, and the quantitative proportioning of air as regulated by fuel-bed resistance alone may be other than that desirable for the most efficient combustion.

As the result of many experiments, metered-air-control systems, which supplement the action of the fuel bed and control the air supplied to a large number of fuel-bed subdivisions on a quantitative basis, have been developed. They are successful both from the standpoint of an operating mechanism and from the results attained.

The installations described provide an appreciable improvement in combustion efficiency under test conditions and a small but measurable improvement in daily operation. Smoking is almost entirely eliminated. Stoker-maintenance expense is reduced and availability is increased. The most important feature is the ability to maintain higher coal-burning rates, which, in some instances, may lead to important reduction in boiler-plant investment.

UNTIL recently, the development of the mechanical stoker has kept up in coal-burning capacity with the growth of the large boiler units that are now quite a common feature of central-station boiler plants. With a few notable exceptions, the choice between stokers and pulverized-coal firing has not been affected by the impossibility of building the stoker large enough. Indeed, the characteristic of the underfeed stoker in the matter of high heat release for a given furnace volume affords a definite advantage in determining the capacity of the unit when space is a serious factor. Whether or not this will continue to be the case will depend in no small degree upon the adequate solution of the problem of air-distribution control with which this paper deals.

In the case of the forced-draft, chain-grate stoker, the necessity for graduating the applied air pressure from front to back, as the blanket of fuel progresses through the processes of combustion, was recognized early in the development of that type, but the advantages of applying the same principle to the control of air supplied to an underfeed stoker were not realized until much later, as will presently be described.

The air-control systems, which incorporate this principle, provide a method to supply and regulate quantitatively the air flow to a large number of subdivisions of the fuel bed. They were originally looked upon only as a means to effect improvement in efficiency of combustion, but development and operation of the equipment have shown that further advantages are possible. The principal ones are important factors leading to increased coal-burning rates and reduced maintenance costs in the operation of underfeed stokers. Today, air-control equipment has become an essential feature to be considered both in the design of new plants and the rehabilitation of old ones.

This paper discusses the air-distribution control problem and describes the methods developed for its solution in some of the most modern stoker plants. The first part reviews briefly the principle of the underfeed stoker as it bears on the problem of combustion control; the second part consists of a short discussion of the control of air distribution by the fuel-bed shape and dampers; the third section describes the modern air-control systems; and the fourth is a discussion of the economics of air control.

I—GENERAL PRINCIPLES OF UNDERFEED STOKER COMBUSTION

The principle of operation of the underfeed stoker, handling bituminous coals of the coking type, depends upon a rather definite physical movement and progress of the fuel on the stoker corresponding to the physical changes which take place during the combustion process. The importance of proper physical movement of the fuel bed, as it affects operating efficiency, cannot be overestimated. Once determined for optimum conditions with any one installation, however, the stoker requires no radical change of adjustment provided the physical and chemical characteristics of the fuel remain fairly uniform.

The path followed in this movement is such that the fuel progresses from the point of introduction as fresh fuel, at the bottom of the retort, to the point of removal as refuse at the surface of the fuel bed. The introduction of fresh fuel crowds along the portions of the fuel bed just ahead, and the movement, continued by the action of successive pushers, at all times is carried on with the minimum of direct contact between the burning fuel and moving parts. Coincident with this movement the fuel passes through zones of increasing temperature below that required for ignition, yet sufficiently high to promote coking with liberation of the volatile matter. Air for combustion discharged into the fuel bed through tuyères mixes with and carries off these combustible gases and their combustion begins, along with that of the coke, as they pass through the incandescent zone at the surface. The coking process tends to close the fissures between adjacent fragments of coal and form a large mass, which, in turn, is subject to some disintegration as a result of action by the pushers. This may be contrasted with the absence of a "closing up" effect in the chain-grate stoker where the blanket of fuel is spread out and transported from front to back but without other disturbance.

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[1] Staff Engineer, Central Heating Department, The Detroit Edison Company. For two years preceding his graduation from Cornell University in 1922 with the degree of mechanical engineer, he was an instructor in the experimental engineering department of Sibley College at that university. Since 1922, a large part of his time has been devoted to the operation of the central heating plants, with particular reference to boiler operation and combustion problems.

[2] Engineer, Production Department, The Detroit Edison Company. Mr. Macomber was graduated from the mechanical engineering course of Michigan State College in 1917. Since that time, with the exception of one year of service in the Chemical Warfare Service, U. S. A., he has been employed by The Detroit Edison Company on problems pertaining to power-plant design and operation.

Contributed by the Fuels Division and presented at the Annual Meeting of The American Society of Mechanical Engineers held in New York, N. Y., December 2 to 6, 1935.

Discussion of the paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1936, for publication at a later date.

Effect of Fuel Bed on Air Flow

In the distribution of air to the fuel bed, the operator is thus dealing with a progressively varying resistance to the passage of air. Consequently, the quantitative proportioning of the air, as regulated by fuel-bed resistance alone, may be other than that desirable for the most efficient combustion.

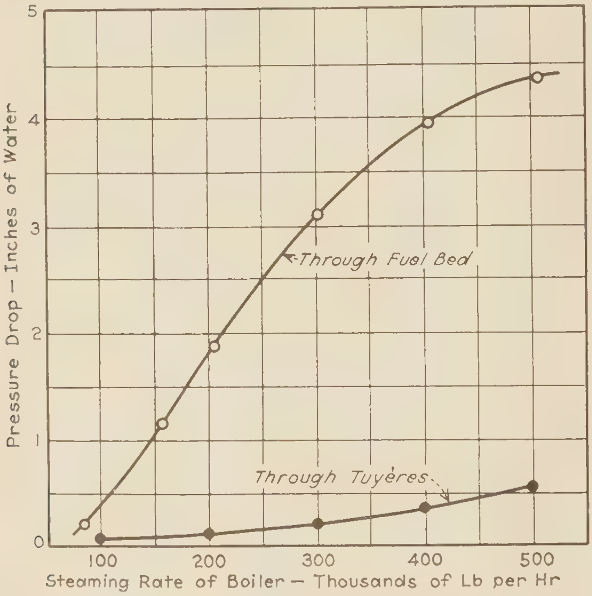

Fig. 1 Pressure-Drop Through Tuyères and Fuel Bed of Beacon Street Heating-Plant Stoker Equipped With Metered-Air Control

In Fig. 1 curves representative of actual operating conditions with an underfeed stoker show that the resistance of the tuyères to the flow of air is of relatively minor importance as compared with that presented by the fuel bed, and is in consequence of little effect in producing a uniform air distribution. The resistance offered by any part of the fuel bed, however, is of major importance. It is in turn affected by conditions such as (1) the shape and thickness, and (2) the porosity as affected by the intermovement within itself, the coking action, and the possible formation of clinkers.

Effect of Air Flow on Burning Rate

The rate of burning coal, expressed in pounds per square foot of projected grate surface per hour, within the operating range, is immediately dependent on the rate of air supply, based, of course, on the presumption that a proper fuel bed has been established and is maintained within reasonable limits of thickness and temperature. Change in the fuel bed to accompany change in coal-burning rate, on the other hand, is not immediately dependent on the rate of feeding coal, but is a cumulative process as compared to the rate of air supply.

Obviously, for a given plenum-chamber pressure the quantitative flow of air, and consequently the rate of burning, will vary inversely as the resistance offered at various points in the fuel bed, which explains why, especially under conditions of forced output, the bed may get accumulatively lighter or heavier. Without some external means of controlling air distribution, thin spots will inevitably develop in some places, while others will become exceedingly heavy. Where the fire is thin, holes will be blown open, causing intensely hot blow-torch effects that may destroy the iron of the grate and induce the formation of clinkers. Eventually, the condition may extend so far that solid particles of coal or coke are blown out of the fuel bed and "drifting" will take place. As a result, the whole fuel bed becomes physically unmanageable and the load can no longer be maintained. Coincident with this condition, an excess of air will blow through in some places while a deficiency exists in others, with an accompanying overall loss in efficiency.

II—METHODS OF REGULATING AIR DISTRIBUTION

The regulation of air distribution for underfeed-stoker operation ordinarily concerns the use of fuel-bed shape, as a resistance to the flow of air, supplemented with a damper system to throttle portions of the air supply as desired.

Use of Fuel-Bed Shape

The use of fuel-bed shape as a factor in regulating the distribution of air relative to the varying stages of combustion from the front to the rear of the stoker has been, and is today in most instances, the only means available for this purpose. Furthermore, in every case its establishment, irrespective of the air-distribution method available, is essential to efficient combustion. It is accomplished entirely by the operator's skill in the use of adjustments to the stoker-operating mechanism as determined by visual observation of the fire. There is a considerable difference of opinion among operating engineers as to what constitutes the optimum physical shape of a fuel bed, and it will not be argued here. In any case, the probable differential in overall economy of the unit will be small.

Use of Dampers

Although the need for a method to supplement the air distribution provided by fuel-bed-shape control was recognized in the early days of stokers, the development of workable damper systems for providing adequate regulation has been slow. Prior to 1910, stokers were small, probably not exceeding 100 sq ft of projected grate area, and, as illustrated by Fig. 2, air was supplied to the entire grate area without any regulation whatsoever between different portions of the fuel bed.

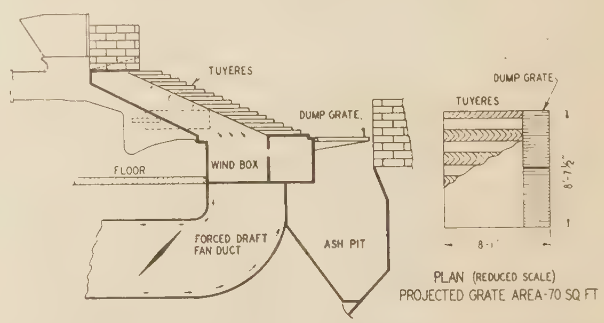

Fig. 2 Early Stoker Installation at Delray Power House No. 1, About 1911

(Air was supplied to the entire fuel bed through one duct which led to a wind box, common to all tuyère boxes.)

With the development of larger stokers it became customary to divide the grate area into several subdivisions and supply air to each through individually controlled dampers. To avoid the complete stoppage of air to any one subdivision, the partitions usually were not tight. A typical installation of this kind in an early plant of The Detroit Edison Company is shown in Fig. 3. The stoker wind box—that part of the stoker to which the tuyère boxes were bolted—was divided into several sections to which the air was admitted through dampers.

On one set of stokers at the same plant further subdivisions

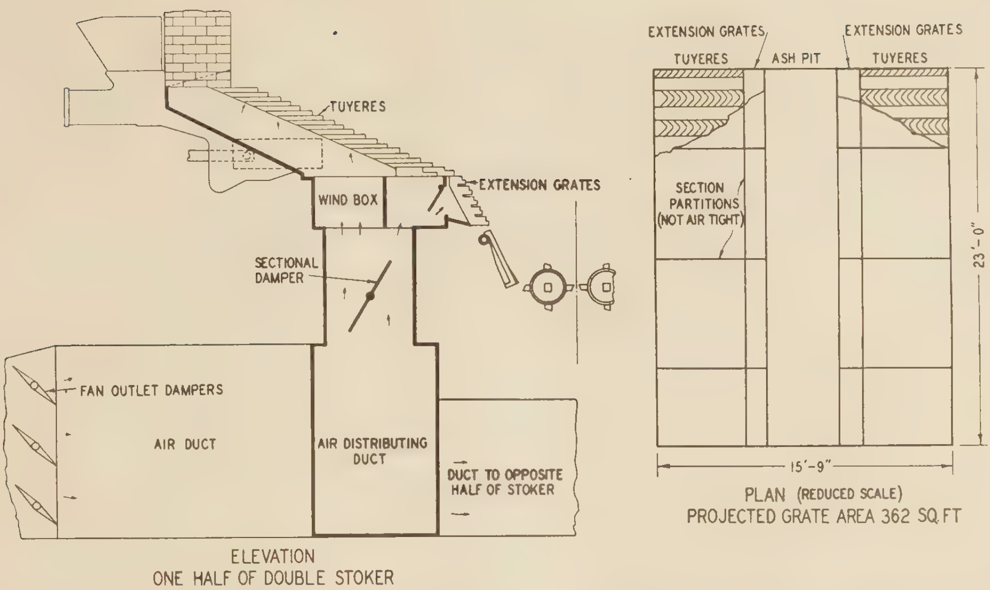

Fig. 3 Sectional Damper System for Stokers of Delray Power House No. 2, Installed 1910–1915

(Each of 13-retort double stokers was divided into four or five sections with separate damper-equipped ducts leading from the air-distribution duct to the stoker wind box. Later the distribution duct was greatly enlarged. The partitions in the stoker windbox were not airtight.)

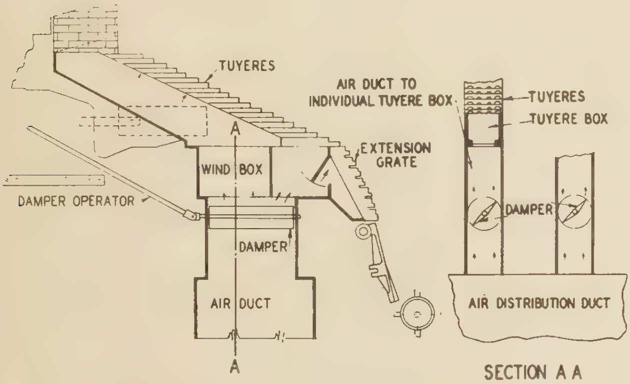

Fig. 4 Experiment in 1911 at Delray Power House No. 2 With Air-Control System for Individual Tuyère Boxes

of the grate area were made during 1911. The wind box was partitioned and hand-operated dampers were installed, as shown in Fig. 4, so that the air to each individual tuyère box could be controlled. The adjustment of these dampers depended upon the condition and shape of the fuel bed as determined by observation. It was concluded, after considerable experimentation, that the multiplicity of dampers thus arranged was of no practical value. Although this experiment was not successful, it was one of the earliest attempts to utilize one feature of the design of the successful air-distribution system reported in this paper, namely, the division of the air flowing through the stoker into a relatively large number of individual streams and the provision of means for controlling them independently.

The use of a plenum chamber was the next step in the development of more desirable air-distribution methods. The high air velocities of the old small air ducts were known to be responsible for much of the localized burning of stoker castings and frequently for the uneven burning of the fuel. The plenum chamber was used, therefore, to provide the more even distribution of air to the tuyère boxes.

This type of construction, which was used in the original Conners Creek installations, is shown in Fig. 5. The tuyère boxes for each stoker, which were divided into several sections, were supplied with air from the plenum chamber through short damper-equipped ducts. This method of air-distribution control became known as "sectional dampering." A section of the fuel bed extended from the front to the rear of the stoker and in width usually included the retorts corresponding to the power-box sections. The dampers were called "section dampers."

A rapid growth in the size of stokers began about 1920 when the first installations were made having 21 tuyères of standard thickness in each stack instead of 17, which had previously been standard. Also, the secondary coal-feeding system was enlarged and in one design consisted of a ram and pusher rather than a single ram. The section dampers for air regulation provided with the older stokers were retained. Other developments, such as preheated air and attempts to increase greatly the coal-burning rates, soon followed. The need for adequate air regulation for the fuel bed became urgent in spite of the mechanical improvements which were made to the stokers.

Stokers having most of the modern features were available about 1926. A typical installation of this period in the Beacon Street Heating Plant in Detroit is shown in Fig. 6. The double stokers, having a projected grate area of 636 sq ft, are 14 retorts wide, have 29 tuyères in each stack, and each retort is fitted with four secondary pushers and a moving extension grate. The air-regulation system for these stokers differs from that used on the older units in that the dampers which control air to the tuyère stacks and to the extension grates are entirely independent of each other. There is no interconnection of air passages above the section dampers, which are greatly improved mechanically.

Since both the tuyère stacks and extension grates of each half of the double stoker are divided into four sections, there is a total of sixteen dampers for regulating the air to the fuel bed. This use of dampers for regulation of air to the extension grates, separate from the tuyère area, is an example of a practice that will be defined as zoning; that is, the subdivision of the total grate area into a number of zone areas, each extending laterally across the stoker from one side to the other.

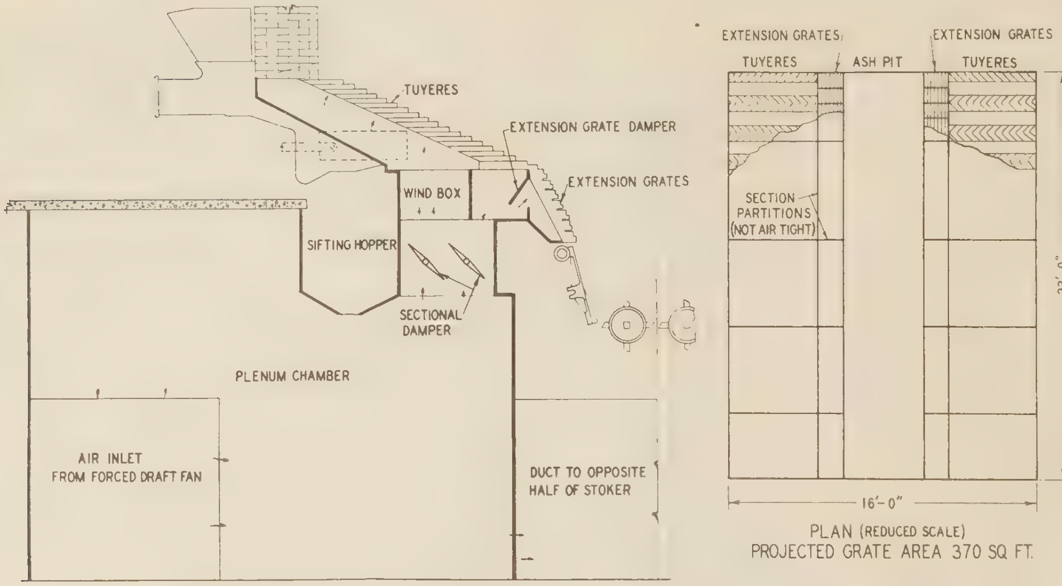

Fig. 5 Sectional Damper System for Stokers of Original Conners Creek Power House
(The divisions of the stoker wind box above the sectional dampers were not airtight.)

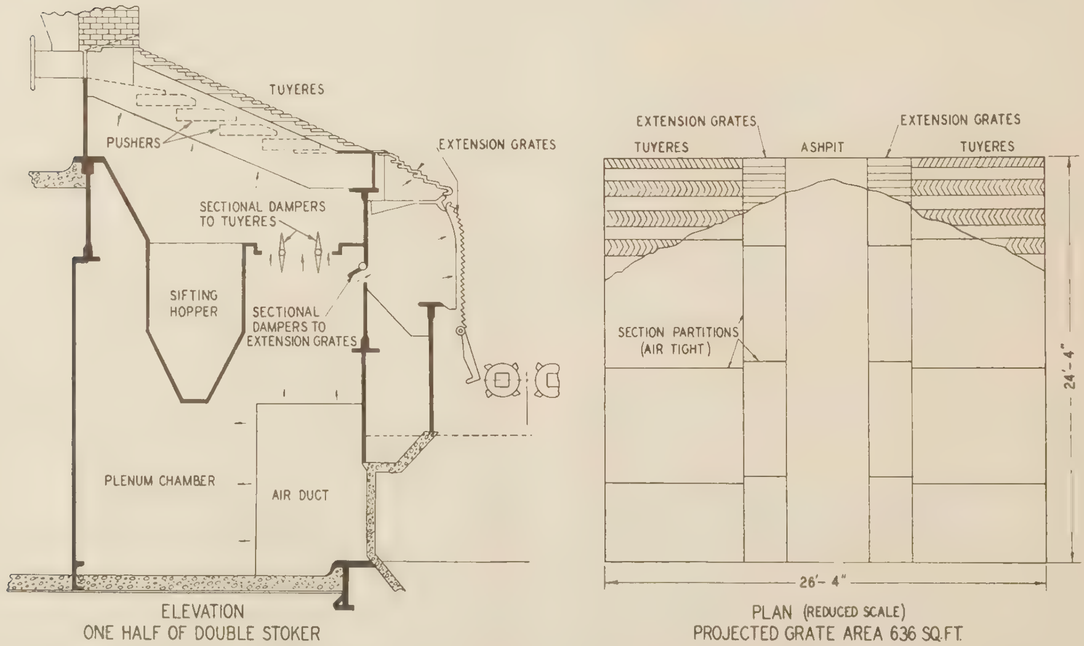

Fig. 6 Sectional Damper System and Fuel-Bed Subdivisions for Original 29-Tuyère Stoker at Beacon Street Heating Plant, Installed 1926
(Airtight partitions divide the tuyère stacks of each stoker into four sections with individual dampers for each section. A similar arrangement is made for the extension grates. The wind box of the earlier stokers is not used with this design.)

In these earlier types of air-distribution systems the fuel beds were divided into a few relatively large areas. The adjustments of the dampers controlling the air flow to the sections were made manually and depended mainly upon observations made of the fire.

It is doubtful whether these systems were of measurable worth, since the boiler operators were much more successful in producing good combustion efficiency by controlling the fuel movement rather than by using the air-regulating dampers. Contemporary attempts were made by a number of other companies to improve the air-regulating methods, and in one case more than forty dampers were installed under a stoker. The results attained, however, were far below expectations.

The shortcomings of the earlier methods pointed out the necessity for dividing the fuel bed into smaller areas and of providing means of controlling the air flow to these areas on a quantitative basis. The efforts to incorporate these requirements led to the development of the present air-control systems.

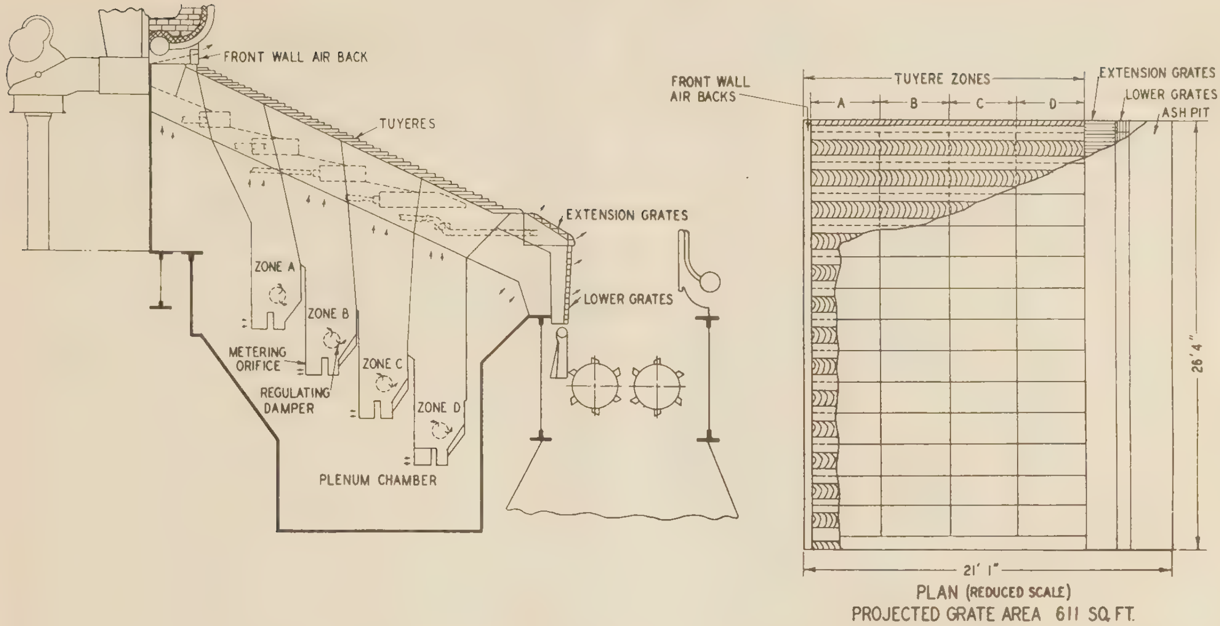

Fig. 7 Automatic Air-Control System and Fuel-Bed Subdivisions of 57-Tuyère, Single-End Stoker for Delray Power House No. 3

(Fifty-six automatic air-regulating units are installed in four zones under the tuyère stacks. No control is provided for air supplied through front-wall air backs and side-wall tuyères. Hand-operated dampers control the air flow to the extension grates. Installed in 1929.)

III—DEVELOPMENT OF AIR-CONTROL SYSTEMS

One of the first schemes for the quantitative control of air supplied to the fuel bed by attempting to compensate automatically for variations in fuel-bed resistance was conceived by Maxwell Alpern. Experimental installations of his apparatus were made at the Edgar Station of The Edison Electric Illuminating Company of Boston in 1925, and at the Marysville Power House of The Detroit Edison Company in 1928. The latter trial installation, which was applied to the original section areas, proved impracticable because of coal dust and siftings, which interfered with the operation of the dampers, and mechanical complications. With the experience gained, however, the system was remodeled and simplified to the form which was eventually installed under the stokers of Delray Power House No. 3.

Automatic Air Control—Delray

The need for an adequate air-control system for the stokers installed at Delray Power House No. 3 in 1929 was apparent, on account of the extremely large size of the fuel bed. These 15-retort, single-end units with 57 tuyères in each stack have a projected grate area of 611 sq ft. Preheated air at 350 F is used. To meet this need, the automatic air-regulation system previously mentioned was installed. It was expected to provide equal quantities of air to each of the fuel-bed subdivisions.

The general scheme of this installation is shown in Fig. 7. The tuyère-stack part of the fuel bed is divided into four approximately equal zone areas, corresponding closely to the portion of the bed controlled by each pusher. Each zone is subdivided according to the tuyère stacks into 14 parts and air is supplied to each of the resulting 56 subdivisions through individual automatically controlled dampers. Air to the side-wall tuyères and the extension grates, approximately 50 per cent of the total for combustion, is not regulated by the automatic-control system. The air to the pit area is adjusted by hand-operated dampers.

Each control unit consists of a duct and two independent cage-type dampers arranged in series. The first damper functions as a variable measuring orifice, and its opening is automatically adjusted with variations of the plenum-chamber pressure. The second damper attempts to maintain automatically a fixed differential pressure across the measuring orifice, irrespective of variation in the fuel-bed resistance over that particular area.

All of the measuring orifices are connected together through a linkage system so that all have the same opening. When the steaming rate is low, the plenum-chamber pressure is low and these orifices are nearly closed. When the steaming rate is high, the increased plenum-chamber pressure moves them to a nearly wide-open position. The differential pressure across any measuring orifice is a function of the quantity of air actually being supplied to the corresponding fuel-bed subdivision, and if the same differential is maintained across all orifices, all subdivisions of the fuel bed should receive equal amounts of air. All regulating dampers, therefore, attempt to maintain automatically the same differential pressure, approximately 1.4 inches of water, across the measuring orifices, irrespective of boiler load. The motive force for the operation of each regulating damper is supplied by a gasometer bell, which is actuated by the differential pressure across the corresponding measuring orifice.

Overcompensation of air distribution in order to maintain approximately equalized resistances across the sections of the fuel bed in a zone area is provided by an iron tube, partially filled with mercury, mounted horizontally on one of the levers of each regulating damper. When the damper is half open, the normal operating position, this tube is level and the mercury is distributed evenly along the tube. If the damper starts to close, the mercury flows toward one end of the tube and closes the damper still further. Thus the air flow to the section involved is reduced below normal. The reverse action takes place when the damper opens more than halfway and the air flow is increased above normal.

After a long period of observation and testing, it has been concluded that although this system has some desirable features, it is lacking in many important respects. Because of its automatic

operation, the control does not place any additional responsibility on the boiler operator and because it regulates the air, to a limited extent at least, fewer manual operations are required in the control of the fire. This is particularly noticeable in the adjustment of the pusher strokes, since the more uniform air flow lessens the formation of clinkers and helps prevent other forms of fuel-bed congestion. This leads to a reduction in the tendency to smoke.

It has been definitely established that both in normal operation and under test conditions higher percentages of CO_2 in the stack gas can be produced when the control is in use. The difference is not large but is a definite indication that the principle of regulating air flow to small sections of the fuel bed is of value in obtaining increased combustion efficiency.

There are, on the other hand, a number of undesirable and limiting features concerning this control system. Because of its bulk, a large plenum chamber is required, and in spite of the precautions which were taken, its operation is hampered by the dust and siftings from the stoker. The initial adjustments of the regulating dampers are difficult to make, and, since the air preheaters of this installation have no bypasses, must be made when the boiler is cold. The action of the dampers in operation can be observed only through portholes in the plenum chambers. Because of the small motive forces available, the equipment has lacked sensitivity, a characteristic now recognized as essential if the overcompensating feature is not incorporated in the control system. Since there is a large pressure drop through the control units, the plenum-chamber pressure is practically double that of a similar stoker without an air-control system. In addition, the control does not report to the operator whether it is functioning correctly in respect to quantity and distribution of air. Neither does it provide means for adjusting the relative air flow between zones. All of these features combine to limit definitely the value of this system.

Manual Metered-Air Control—Beacon Street

The motive underlying the installation of a manually operated metered-air-control system at the Beacon Street Heating Plant in 1929 was a desire to improve combustion efficiency if possible and to study the air-distribution requirements of a large stoker. At that time three boilers equipped with 14-retort, double-ended stokers having a projected area of 636 sq ft including the ashpit had been installed and were operating in an acceptable manner. A fourth boiler was to be installed the following year. A careful analysis disclosed that while the average combustion efficiency based upon stack-gas analyses was good, there were many periods during which better results should have been obtained. Unevenness of the fuel bed usually appeared to be the cause of this inferior operation. It was decided, therefore, to install an air-control system that, although concurrent with the Delray installation, would be less complicated and yet would permit of a greater flexibility and knowledge of air-distribution requirements. In the resulting installation, the air supplied to each fuel-bed subdivision was metered and the dampers were adjusted manually rather than automatically. Because of the success attained by this installation, the old sectional air-regulation system of another boiler was replaced by an improved hand-operated, metered-air-control system in 1931.

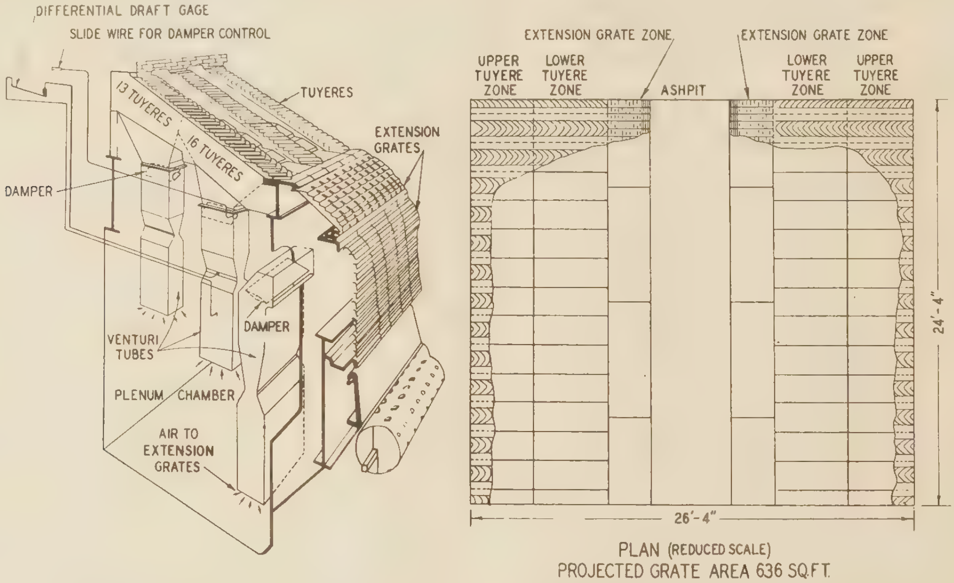

Fig. 8 Manual Metered-Air-Control System and Fuel-Bed Subdivisions at Beacon Street Heating Plant
(The tuyère stacks are divided into two zones while the extension grates are the third zone. Air is supplied to the fuel bed through 68 individual venturi tubes, each equipped with differential gages and hand-controlled dampers.)

This control system, together with the subdivisions of the fuel bed, is shown in Fig. 8. The tuyère stacks proper, because they are shorter than those of the Delray single-ended stoker, are divided into only two zones; the upper comprising 13 tuyères and the lower 16 tuyères. Air is supplied to each tuyère zone through 15 adjustable-throat venturi tubes, one for each tuyère stack. The air flow in each is regulated by a damper installed in the duct above the venturi throat. A sliding wire in a flexible tube extending to the boiler gage board is used to operate the damper, and the position of the control button on the end of the wire indicates the position of the damper. Inclined differential draft gages connected to the venturi tubes are used to indicate the flow of air. The third or extension-grate zone for each of the stokers, because of structural difficulties, was divided into only four sections. The air to each is metered and controlled in a manner similar to that used for the tuyère zones. Thus, for this installation, the fuel bed is subdivided into 68 separate areas and as far as possible, the control system is applied to all air supplied for

combustion. It does not, however, control air which enters the fuel bed through small openings around the secondary coal pushers and other similar locations, which may amount to as much as 20 per cent of all the air required.

In order to obtain recognizable venturi differentials at low air-flow rates and at the same time minimize the draft loss at high air-flow rates, the venturi throats were made adjustable. They are so constructed that the maximum boiler load can be carried with a differential of one inch of water, of which approximately 40 per cent is an unrecoverable loss.

For normal fuel-bed conditions the same amount of air is required for all sections of one zone and the dampers should be in the same relative position—approximately half open. The boiler operator can then close the dampers sufficiently to maintain average flow in those sections where the air flow tends to be high or open those where the flow is low. With this manual control, it is possible to reduce the air flow in any section which tends to burn light to an amount below the average for heavy spots. This overcompensation undoubtedly could be made to accelerate the correction of an uneven fire; but in actual practice the operator is instructed to manipulate the dampers periodically in an effort to maintain uniform flow in all sections of a given zone. To guide the operator in maintaining the proper distribution of air between zones, scales on the air-flow gages indicate the proper flow for different boiler loads. The overall combustion results are indicated by the CO_2 recorder.

The frequency with which the dampers must be adjusted depends upon many different conditions. For low fuel-burning rates, adjustments may be required only three or four times an hour. For very high combustion rates, however, rather frequent adjustments are required to part of the dampers at least, on account of the high air velocities through the fuel bed and the tendency to drifting.

The efficiency of combustion, already rather high except under certain conditions as previously explained, was not greatly improved by the air-control system, although under test conditions it was possible to maintain appreciably higher CO_2 in the stack gas, particularly with the low fuel-burning rates. Curves illustrating the CO_2 produced with and without the air-control system are shown in Fig. 9. With the higher fuel-burning rates, the improvement in CO_2 with the control in use is due to the more even fire which is maintained and to a better air distribution between the upper and lower tuyères. The admission of unnecessary excess air to the combustion chamber through thin spots in the fuel bed is considerably reduced.

The improvement in CO_2 as a result of the control at the lower fuel-burning rates has a somewhat different explanation. With the decrease in boiler output the lower end of the fire thins out until there is very little active combustion over the extension grates and lower tuyères. This reduces the resistance to air flow and permits large quantities of air to enter the combustion chamber when there is no control. While adequate dampers had originally been provided for the extension grates, there was no method of gaging the effectiveness of their use and the operators were reluctant to close them, because of the danger of burning castings. Also, there was no method of controlling the air flow through the lower part of the tuyère stacks. With the control system, the differential gages continuously indicate the air-flow rate, and thus it is possible to reduce the air flow to the extension grates without danger of destroying stoker iron by a complete shut-off of the air. The dampers for the lower zone of the tuyères can be partly closed until they pass only sufficient air to burn the fuel efficiently. This particular feature quite clearly demonstrates the need for quantitative control to make any air-regulation system effective.

Although the improvement in combustion attributed to the control system appears definite from a test standpoint, a corresponding improvement in the overall plant performance has not yet been demonstrated. During one winter the two boilers equipped with the air-control system and the two with the original section-damper system were operated for alternate two-week periods. Operating records indicated that the boiler efficiency was 0.3 of a point higher on the percentage scale when the metered control was used—a difference which is not sufficiently large to be significant. Under these conditions the control adjustments were a part of the fireman's routine. The failure to obtain as much improvement as the tests had shown possible suggested

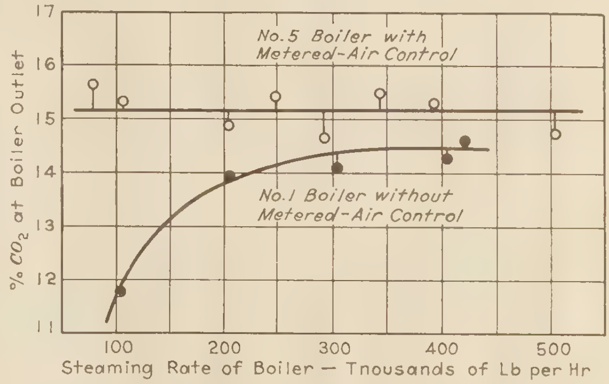

Fig. 9 CO_2 Produced by Beacon Street Boilers With and Without Metered-Air-Control Systems
(Data from tests of 8–16 hr duration.)

that the fireman failed to use the dampers to the best advantage and hence confirmed the desirability of automatic and substantially instantaneous adjustment of the dampers—a feature incorporated in an ensuing installation at Conners Creek.

Although it was not anticipated at the time the Beacon Street installation was made, the air-control system has made possible a large increase in the maximum dependable fuel-burning rate of the stoker. With the stokers as originally installed, the boilers had a four-hour maximum steaming rate of 435,000 lb per hr, which corresponds to a coal-burning rate for the stoker of 63 lb per hr per sq ft. With the air control, the conditions are so improved that the dependable four-hour maximum output is 535,000 lb of steam per hr while the coal-burning rate for the stoker is 78 lb per hr per sq ft. This is a 23 per cent increase in steaming rate, and when all four boilers in the plant are equipped for metered-air control, they will be able to serve a load which originally would have required five boilers.

The limit of dependable coal-burning capacity, either with or without an air-control system, is determined by the ability to keep the fuel bed in a workable condition. As previously explained, with the high air velocities occurring when operating at high combustion rates, there is a tendency to lift small particles of coke from the fuel bed and deposit them in drifts in other parts of the fire. This action, if permitted to become sufficiently aggravated, eventually causes whole sections of the fuel bed to become thin, while heavy drifts, through which no air will pass, make their appearance in other sections. The effect is cumulative, and it is soon necessary to reduce load until the fuel bed can be worked into proper shape and condition. Without an air-control system, the fuel-burning rate must be kept sufficiently low to prevent this drifting from becoming serious.

With this manually operated, metered-air system, however, the distribution of air is maintained more uniformly and the runaway effect just described can be controlled to permit a 23 per cent increase in coal-burning rate. The elimination of thin spots,

with their resulting high air velocities, minimizes the tendency for the small particles of coke to drift. At high combustion rates, because of the high flame concentration, mere observation of the fire does not reveal the presence of thick or thin spots in the fuel bed or congestion due to clinkers. The air-flow gages, however, together with the damper positions, indicate these defects to the operator, so that the correct adjustment of the rate of coal feed and, if necessary, of the pusher travel can be made. It is possible, therefore, to attain the higher continuous fuel-burning rate without danger of disrupting the fuel bed.

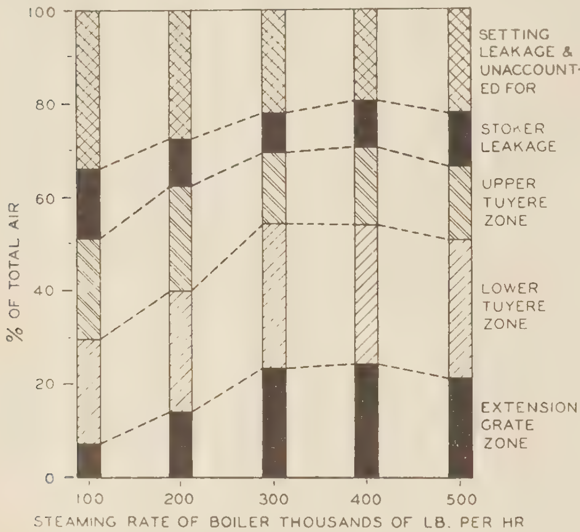

Fig. 10 Optimum Distribution of Air to Beacon Street Heating-Plant Stoker Equipped With Metered-Air-Control System
(Total air calculated from analyses of gas samples taken at boiler outlet.)

Metering the air, as made possible by this system, permits a more detailed study of the effects of varying air distribution on combustion efficiency. With the dampers provided, the relative amounts of air supplied to each of the three zones may be increased or decreased. In a number of constant-load tests to discover the optimum distribution, the portion of air to the upper zone was reduced to a minimum and then progressively increased to the maximum possible. Corresponding changes in the opposite direction were made in the portion of air supplied to the lower tuyère zone while sufficient air to complete the burning of the coal was supplied to the extension-grate zone. Combustion results were judged by Orsat analyses of the stack gas, and the optimum air distribution thus determined.

The distribution at the different steaming rates which produced the best combustion results is shown in Fig. 10. To maintain these conditions, the amount of air supplied the upper zone was reduced by partly closing the dampers appreciably below that which would have obtained without a damper system. Too large a reduction, however, caused severe smoking. It should be noted that this particular proportioning applies only to stokers using cold air similar to those on which the tests were made and may depend also on the pusher stroke adjustments. A comparison of results of these tests with those obtained where preheated air is used indicates that due to the increased ignition rate, which preheated air provides, the optimum distribution of air may be different.

In other tests, attempts were made to improve combustion by altering the length of stroke of the secondary coal pushers. It was found, however, that the length of pusher strokes which had been determined before the air control was installed were the best that could be had. Any large changes resulted in smoky stacks.

Aside from the combustion considerations, this air-control system is of measurable value in reducing the cost of stoker maintenance, as will be shown later in this paper. When in use, a more uniform air flow is maintained through all sections of the fuel bed and localized burning of stoker castings is reduced.

The use of this type of control, then, has definite advantages to offer in the design and operation of a plant. Principally, it permits increased steaming rates and lowered maintenance charges for the installation to which it is applied. A somewhat minor gain is possible in combustion efficiency.

On the other hand the system has certain disadvantages, but considering the results achieved, none of them is serious. The venturi tubes and dampers necessarily take up a large portion of the space in the plenum chamber but the under parts of the stokers are still sufficiently accessible to permit repairs. Mechanical difficulties of a minor nature are encountered occasionally.

Readjustment of the dampers to bring the air-flow gages in line is not so formidable a task as it appears, since ordinarily only a few require adjustment at one time. At the high coal-burning rates, however, an unbalanced fuel bed may develop quite rapidly and consequently more frequent attention is required. Observers have estimated that ordinarily no more men will be required to operate the boilers equipped with this type of air control up to their full steaming capacity than is required for the boilers with no control. In the event that additional labor is required, the added cost is far outweighed by the saving in fixed charges on the plant investment resulting from the additional steam-generating capacity provided by the control.

The possible advantages to operation which result from the use of this control were further demonstrated by subsequent installations at the Hudson Avenue station of the Brooklyn Edison Company, where an average coal-burning rate of 111 lb per sq ft of projected grate area per hr was maintained for a four-hour period. It was observed here that the attention required for manual control under these conditions might limit its use. This factor, together with that which had previously been demonstrated in the Beacon Street tests concerning the effect on combustion efficiency resulting from continuous attention to damper regulation, led to the decision to make the control for a new Conners Creek installation partially automatic, and instantaneous in operation.

Automatic Metered-Air Control—Conners Creek

In the rebuilding of the Conners Creek power house, metered-air control, for the first time, was an important consideration in the design of a large central station. The plant which originally had a generating capacity of 180,000 kw served by 14 boilers is being reconstructed, using new boilers for higher pressure and superheat to serve eventually 330,000 kw of generating capacity.

One of the major problems in the design of the new boilers was that of providing adequate steam-generating capacity for the greatly increased electrical output. Although some increase in grate surface was possible, the location of the building columns definitely limited the size of the new stokers. The installation of additional heat-absorbing surface was possible by increasing the height of the boiler house, by installing air preheaters and economizers, and by the addition of highly effective surface in the screen tubes forming all four walls of the new furnaces. It was necessary, therefore, either to use a method of increasing the coal-burning capacity of the stokers or to install a larger number of boilers of less capacity.

During the period when this rebuilding program was being studied, the air-control installations at Beacon Street were being tested and the ability of this equipment to increase the maximum

dependable coal-burning rate was definitely established. It was decided, therefore, to equip the new stokers with air-control systems and thus obtain a greater rate of heat release than otherwise would have been practicable. With the stokers equipped in this manner, each new steam-generating unit was designed to have a two-hour, maximum capacity of 420,000 lb of steam per hr as compared with 325,000 lb per hr which would have been the extreme limit without air control. Present indications are that this 420,000 lb per hr steaming rate can be maintained continuously. The old boilers, occupying the same floor area, had a continuous steaming capacity of 125,000 lb per hr. A total of 12 new boilers, because of the use of control, instead of a possible 14 without control, will be sufficient for the completed plant.

The air-control system used in these installations has meters and dampers resembling those at Beacon Street, but the dampers are operated by hydraulic-power cylinders controlled from diaphragm-type regulators. The general scheme of the installation is shown in Fig. 11. The grate surface of the stoker is divided into three zones. Although the tuyère stacks are of the same height as those at Beacon Street, thinner tuyères are used. There are 23 tuyères in the top zone while the remaining 18 tuyères are in the middle zone. The lowest zone consists of the reciprocating extension grates. The stoker is 12 retorts wide, and the two stokers serving each boiler have a total projected grate area of 557 sq ft.

The division of the grate area differs from that which had previously been used in that the tuyères are center-divided so that the tuyère stacks are partitioned down the middle. Each venturi tube furnishes air to the half of the tuyère stack on adjacent sides of each retort. This makes each air-control subdivision a unit corresponding to the coal retort. Also, the number of extension-grate sections, each with separate venturi tube and control damper, was made equal to the number of retorts. Thus for the double, 12-retort stoker, a total of 72 meters and control dampers are required for the six zones.

The regulators chosen for this rather exacting service are sensitive to such infinitesimal variations in differential pressure as to obviate the necessity for contracting the throat at light loads. Their primary function is to maintain a constant flow of air to the area under control, irrespective of change in fuel-bed resistance. It is in this respect that this control differs in operating principle from the automatic overcompensating design at Delray, or that possible in practice with the Beacon Street or Hudson Avenue installations. All regulators of one zone have the same simultaneous adjustment through loading-pressure diaphragms and therefore equalized air flow is maintained continuously through all meters.

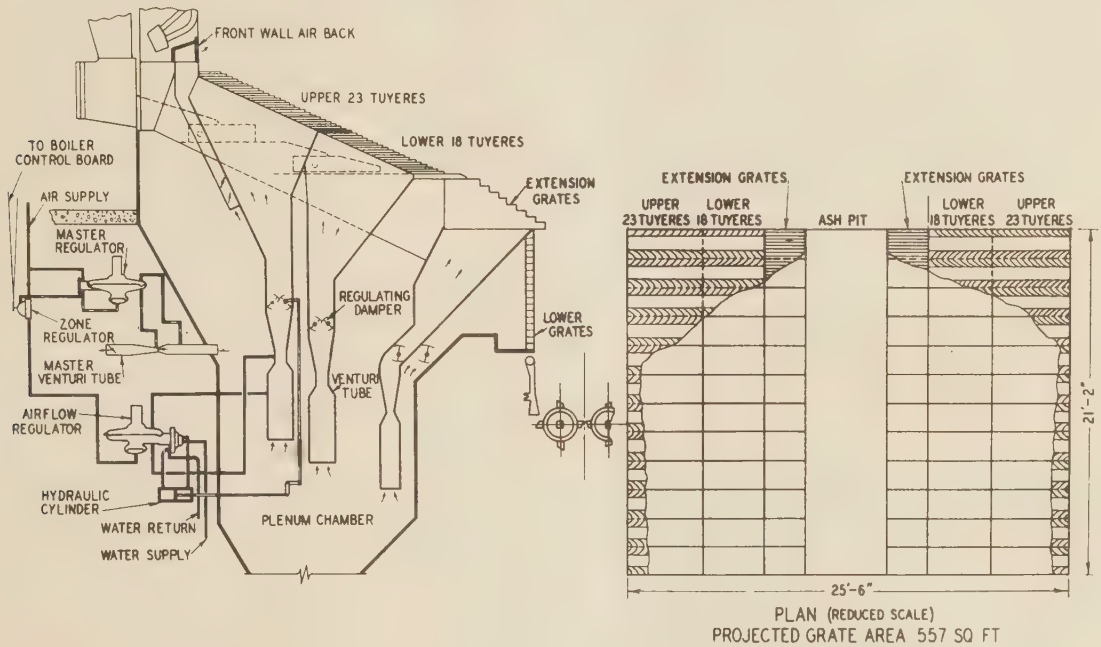

Fig. 11 Automatic Metered-Air-Control System and Fuel-Bed Subdivisions for New Conners Creek Power House Double 12-Retort Stokers

(Center-partitioned tuyères are used and the air-control area is from center of tuyère stack to center of tuyère stack.)

Adjustment of distribution between zones is accomplished by six zone-loading regulators, each one of which acts in conjunction with all 12 air-flow regulators of one zone. By adjusting these zone regulators, the amount of air which will flow through the meters of one zone with respect to another can be changed to meet the requirements of the fire. To readjust the air flow automatically with change in load, the zone regulators are in part subject to air-loading adjustment through a master regulator which is influenced indirectly by variations in plenum-chamber pressure. To change air-flow rates would otherwise require manual readjustment of the six individual zone regulators.

One set of 12 and one set of six air-flow gages are provided for each boiler. A gang valve permits the connection of the twelve-gage set to all of the venturi tubes in any one selected zone. One unit of the six-gage set is connected to a key venturi in each of the six zones. With this combination it is possible to check the action of all the regulators. It has been found that except when the dampers are allowed to reach the limit of their travel, the regulators maintain a uniform air flow through all venturis in a zone. The six-gage unit indicates continuously the relative distribution of air between zones. As the reliability of the regulators is established over a period of time, it is anticipated that the 12-gage unit and piping may be eliminated.

The first boiler equipped with this control system went into operation in November, 1934, a second was completed a short time later, while a third was commissioned in August of this year. The air-control system of the third boiler differs from the first two in that damper-position indicators are also provided at the control board to give information of relative fuel-bed resistance.

Tests during the first year's operation have definitely established that, with the control system, the stokers are capable of attaining the coal-burning rate which was expected. Operating performance tests of four hours' duration have been made at steaming rates up to 425,000 lb per hr. The coal-burning rate at this load is 75 lb per sq ft of projected grate surface per hr. The excess air in the furnace was 12 per cent, and there was neither measurable combustible in the gases nor smoke at the end of the superheater pass.

It is felt that this control is a distinct step forward in the development over the manually operated type. It has retained the basic features of the latter and has provided others that are considered desirable. From the operating standpoint it is possible to maintain maximum burning rates under conditions of high efficiency. Smoking is almost entirely eliminated under all conditions of operation. Attention by the operator to adjustment of the control and the stoker is, under normal conditions, greatly reduced. Because of the increased reliability of the stoker, the continuous operating period between shutdowns is greatly lengthened and maintenance costs are correspondingly lessened.

IV—ECONOMICS OF METERED-AIR CONTROL

In considering the economic justification for metered-air control, it should be remembered that this form of control does not replace the usual forms of automatic combustion control which adjust the ratios of total air and total fuel in proportion to the load demands. Rather, it should be regarded as a supplement thereto. It delivers to each subdivision of the fuel bed its proper quota of the total air supply. At Beacon Street, which is equipped with a well-known system of combustion control, the metered-air control serves as a refinement of and in no way an interference to the former.

Justification of the investment in metered-air control must result from a consideration of the following items: Increased efficiency of combustion, decreased maintenance cost of stoker, increased capacity of the unit, intangible items, and cost of control, including operating expense. Four of these items represent benefits which will accrue from the use of the control, while the fifth represents the added cost.

Combustion Efficiency

Increased efficiency of combustion is to be expected because of the more accurate adjustment of the fuel-air ratio in each fuel-bed subdivision. Test results show a marked improvement, particularly at the lower steaming rates. The only overall, long-time test conducted to date at Beacon Street showed an improvement of 0.3 per cent. While this appears low in view of the test results, a larger differential is possible with use of the Conners Creek type of control, because of continuous and instantaneous automatic adjustment of the dampers. However, no greater improvement should be credited unless and until demonstrated by experience.

Maintenance Expense

The most important item of expense in the operation of a boiler plant after fuel and operating labor, is stoker maintenance. Often more can be saved by efforts directed toward minimizing it than in attaining higher combustion efficiency. It is an expense subject to many variables for a particular installation. Among these are the coal-burning rate, the accumulated total of hours in service, the temperature of the air supply, the grade of coal as well as the skill and attention of the operator. Some of these factors are obviously determined by plant design and others are to an extent controllable in operation.

In the Beacon Street Heating Plant a record of stoker maintenance costs over a period of years of operation both before and after the installation of metered-air control, presented in Table 1, shows for one boiler a differential of 4.2 cents per ton of coal

TABLE 1 COST OF MATERIAL USED FOR STOKER MAINTENANCE—BEACON STREET HEATING PLANT

No. 5 Boiler

Year	Coal burned, tons	Stoker maintenance material: Total, dollars	Stoker maintenance material: Per ton of coal, cents
Prior to metered-air-control installation			
1927	6,510	292.14	
1928	21,370	2135.19	
1929	26,500	1062.69	
1930	20,700	1950.02	
	75,080	5440.04	7.2
After metered-air-control installation			
1932	18,400	340.81	
1933	12,600	432.00	
1934	30,438	1072.98	
	61,438	1845.79	3.0

Entire Plant, Sept. 1, 1926, to Dec. 31, 1934[a]

	Coal burned, tons	Stoker maintenance material: Total, dollars	Stoker maintenance material: Per ton of coal, cents
Without air control			
Boiler 1	145,097	8079.35	5.6
Boiler 2	169,566	7882.41	4.6
Boiler 5	75,080	5440.04	7.2
	389,743	21,401.80	5.5
With air control			
Boiler 4	98,509	3929.32	4.0
Boiler 5	61,438	1845.79	3.0
	159,947	5775.11	3.6

[a] Year 1931 for No. 5 boiler omitted, as air-control system was installed during the year.

burned in favor of the control. The differential for the entire plant is not so great as this, but both boilers equipped with air-control systems have a lower cost of stoker-maintenance materials than any boiler without the air control. As already mentioned, this plant is without air preheaters.

At the Delray No. 3 Power House, initially placed in operation in 1929 and equipped with preheaters for a combustion-air temperature of 350 F, slightly lower than that at Conners Creek, the cost of maintenance material for the years 1933 and 1934 was 10.5 cents per ton of coal burned. At Conners Creek the new boilers were first operated in November, 1934. Although the length of time these stokers have been in operation is too short to establish the average maintenance cost, it is the authors' opinion that the air control will account for a reduction of about five cents per ton below that which would have obtained without the control. This item evaluated on the basis of 60,000 tons per year coal consumption per boiler would amount to $3000.

The point might be raised that a comparison of the Delray and Conners Creek installations does not illustrate fairly the advantage attributable to air control for the reason that both have it. Delray No. 3 does have control, but of a type and design that, although affording some advantages, has also many deficiencies compared with the later Conners Creek installation. It is, in fact, now planned to substitute air-control equipment of the Conners Creek type for that originally installed.

Capacity

Tests before and after installation show, conservatively, that the system is capable of increasing the maximum output by at

least 23 per cent. Actually, the control goes farther in its effect on output in that the maximum continuous steaming rate is caused to approach the two-hour maximum.

A discussion of this item of capacity raises the question whether or not this increase in capacity is obtained at less cost than if attained through installation of additional grate area; whether, in other words, it is more profitable to increase the burning rate on a given grate area or provide more grate.

While the answer to the question may not be obvious in the design of a new plant of which the floor dimensions are not yet fixed, the advantages are more evident when it is a case simply of greater utilization of existing space. In considering rehabilitation projects, the floor area as limited by column spacing is, of course, an important factor, and the use of an air-control system permits the installation of fuel-burning equipment of a capacity which otherwise would be impracticable. An actual example of this sort of problem was involved in The Detroit Edison Company's rebuilding program at the Conners Creek Power House. A successful solution of that problem was made possible largely by the fact that the coal-burning rate on an amount of grate definitely limited by existing column spacing could be increased sufficiently to balance the greatly increased heat-absorbing surface made possible by the greater height of the rebuilt boiler house.

Because the air-control system has stepped up the capacity of the fuel-burning equipment, the necessity of installing the thirteenth and fourteenth boilers for the completed plant will be eliminated. Thus the cost of 12 air-control systems at $20,000 per boiler, totaling $240,000, will offset an otherwise possible investment of $1,500,000.

The returns on the investment through increased efficiency and reduced maintenance are necessarily dependent on the period of operation and the output of the unit. Savings in capital investment, on the other hand, are in effect continuously throughout the life of the equipment.

Intangible Items

There are certain other intangible items to be credited, but which are not readily evaluated.

Availability. At the time of writing this paper six months have passed without the need for shut down for repairs to the stokers at Conners Creek. Necessarily, any influence which the use of this control may have toward extension of the period between shutdowns must depend upon the reduction of stoker iron burnouts. Although burnouts may occur during periods of starting and prolonged banking, they are most frequent with high burning rates. Beyond burning rates of 70 lb per sq ft per hr, other limitations, such as wall slagging, front-tube and superheater slagging with their attendant and sometimes cumulative effects on superheat temperature and drafts, can impose a more effective obstacle to further increase of the length of operating period than does the stoker itself.

Smoke and Fly-Ash Nuisance. It is apparent at Conners Creek that use of the air-control system on the new boilers almost entirely eliminates smoking under all conditions of operation. Furthermore, the practically uniform distribution of air through the fuel bed and the avoidance of "blowtorch" action appears to reduce the amount of cinders carried over into the boiler passes and up the stack.

Operating Attention. The amount of attention involved in the operation of a stoker with and without control is dependent upon the type of control and the coal-burning rate. Without control, there is the constant need of observation of the fire through inspection doors and adjustments of section dampers, pusher stroke, and speed of stoker sections. Instead, with the manually operated control, the operator reads the air-flow indicating gages and adjusts the meter dampers accordingly for approximately equal distribution within a zone and the proper proportioning between zones. With the Conners Creek control, the individual adjustments being automatically and instantaneously corrected, only manual readjustment of proportioning between zones is left to the operator, thus relieving him of much mechanical routine and attention to detail. At Conners Creek under the peculiar operating conditions existing at present, with only two boilers in service, it has been advisable to have one operator per boiler. As more boilers are put in service and operators are trained in the use of the air control, it is expected that one operator will be able to take care of two units.

Cost of Control

The cost of metered-air-control equipment depends upon the type of control, the multiplicity of units involved, and the number of accessories, such as instruments and their incidental piping. Generally, in the initial installation there is an overrefinement in this respect, which is then followed by simplification.

The introduction of position indicators for the automatic dampers at Conners Creek has been found a necessity from the standpoint of fuel-bed control. They will permit the elimination of all air-flow gages and related piping except one per zone, which will be used when adjusting air distribution between zones. Further simplification to lessen the attention of the operator is being attempted with a rearrangement of master control devices for zone apportioning. These changes, however, will have little effect on the cost of the complete equipment which now stands at $20,000 per boiler.

When an additional installation is made at Beacon Street, the design probably will be simplified by using curved tube air-flow gages which will permit the elimination of the variable-throat-area feature of the venturi tubes. The cost of such an installation is estimated as $12,500.

Operating cost for power to supply the Conners Creek automatic equipment with compressed air and water is negligible. The required air supply per boiler is 1 cfm at 8 lb per sq in., while the water supply required is 35 gpm at 50 lb per sq in.

Fan power for equivalent air quantity, as represented in excess plenum-chamber pressure, is necessarily more with control than without control. With control, the forced-draft fan must deliver to the plenum chamber, for any specific boiler load, the quantity of air required at a pressure dictated by the maximum resistance of the fuel bed plus that of the control metering boxes and dampers. All the dampers throttle the flow, but to a varying degree, depending upon zone flow proportioning. Without control, the resistance is the average for the fuel bed, the air seeking the path of least resistance. Experience, however, seems to point to the fact that there is very little difference in the total power requirements for forced- and induced-draft fans, either with or without control, of the Conners Creek or Beacon Street types. The reason for this is that with control the extra plenum-chamber pressure required is largely offset by a lower excess-air ratio.

CONCLUSIONS

The use of either manual or automatic metered control for the distribution of air supplied to large underfeed stokers is justified in present-day operation, although the automatic metered control accomplishes more as it adjusts the air flow instantaneously and continuously, whereas the manual metered control depends upon the operator's observing the air-flow gages and making the necessary adjustments.

The reasons that justify the use of metered-air control are:

(1) It increases, by twenty-three per cent or more, the maximum attainable combustion rate of the stoker which can be carried successfully.

(2) Stoker maintenance cost is appreciably reduced through the reduction of burnouts.

(3) The availability of the unit is correspondingly improved because of less frequent shutdown for repair.

(4) By reducing the excess air, it gives an improvement in efficiency, particularly at the lower fuel-burning rates.

(5) Smoke is almost entirely eliminated under all conditions of operation.

(6) Because of the increased fuel-burning rate made possible, a smaller stoker can be used, which in some cases leads to marked saving in the entire boiler-plant investment.

Although a capable operator is still required, use of the control does relieve him of many duties which otherwise would be necessary, since any changes in the fuel bed are at once visibly indicated and can be readily corrected before the conditions become acute.

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Film-Lubrication Theory and Engine-Bearing Design

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In this paper the data of film-lubrication theory have been adapted to the purposes of bearing design for reciprocating engines, particularly for the case of the high-speed internal-combustion engine. Suitable corrections for end leakage in short bearings have been incorporated and the results are presented in the form of design charts. Some published experimental data have been correlated with reference to the theory. The flow of oil in pressure-fed engine bearings has also been treated, the oil being considered as a coolant as well as a lubricant. The paper includes a brief discussion of the application of the data in design, with remarks on some detail features and practices.

THERE are in existence carefully constructed charts which present the results of film-lubrication theory in a form suitable for many purposes of design. As a rule these charts were prepared with a view to their usefulness in the calculation of bearings for rotary machinery such as turbines or generators. There is no doubt that film theory is more generally applied in that type of design than in the case of reciprocating-engine bearings. This may be explained in part by the fact that some peculiarities of engine bearings were not provided for in the construction of the charts, with the result that their proper application is not readily apparent.

The object of the present paper is to adapt the existing results of film theory to the design of engine bearings, especially those of high-speed internal-combustion engines. An effort has been made to present the data in a form suitable for general use. In conjunction with the theory, consideration has been given to special aspects such as cooling.

Published experimental data for bearings similar to those considered can readily be compared with the theory. This has been done, with two objects: First, to illustrate the extent of variation within the region of true film lubrication, and second, to estimate if possible the limit of loading, above which film theory can no longer be expected to apply.

Some suggestions have also been added on the application of the design data in practice.

The subject of "boundary" lubrication is excluded from consideration, together with the related questions of oiliness of lubricants and the relative merits of various metal surfaces for bearing purposes. The viewpoint is that bearings should be designed from the standpoint of film lubrication, and that a capacity to withstand partial metallic contact without serious injury should be counted as a safeguard rather than as a normal requirement.

Type of Bearing To Be Considered

The bearing to be considered in this paper is typical of the main and crankpin bearings of an internal-combustion engine. Its special characteristics are assumed to be as follows:

1 In structure it is of the complete type. Both bearing and journal are substantially bodies of revolution, differing in diameter by a small clearance. Such a bearing is equally capable of withstanding a force acting in any direction normal to its axis. In fact, forces acting on engine bearings fluctuate rapidly both in magnitude and in direction. It does not follow from the bearing structure that its pressure distribution will be that of the classical 360-deg bearing, and it is known that there is no correspondence in this respect.

2 The length-diameter ratio may be small, for example, 0.5 or even less.

3 In some cases the bearing may operate with a high degree of eccentricity.

4 The rate of heat generation is sufficiently high so that the action of the oil as a coolant, as well as a lubricant, must be considered.

5 The oil is supplied under pressure in such fashion that the entire clearance space may be considered as filled with oil at all times. Since the oil is normally supplied through grooves or holes near the longitudinal centerline, its path of flow to the point of escape is generally axial.

Several of these particulars render this type quite distinct from the common partial bearing designed to carry a load acting constantly in one direction. Account is taken of its peculiarities in the following discussion of the theory.

Complete Bearing With Forced Axial Lubrication

Fig. 1 illustrates diagrammatically the elements of the bearing. Oil is assumed to be delivered under pressure to a groove running entirely around the center of the bearing. The dimensions of the groove are such that there is no noticeable pressure variation within it when passing around the circumference. As already mentioned, it is assumed that the supply pressure, and the corresponding rate of flow, are such that the clearance will be maintained full of oil. However, the dynamic load-supporting pressures are assumed to be unaffected by supply pressure.

It is apparent that the whole bearing will behave like two shorter bearings of length L, each supporting a load F equal to one half the total load. The discussion will refer to one of these sections of length. If it were practicable to dispense with the central groove, the carrying capacity obviously might be increased greatly. Later in the paper, alternative methods of pressure feed will be considered.

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The journal may first be pictured as rotating at a given speed under a condition of zero load, that is, concentrically within its bearing. Assuming oil supply and other conditions as already stated, the case is then characterized by:

1 An applied pressure per unit of projected area P, such that $F = PLD$. For the initial condition, $P = 0$.

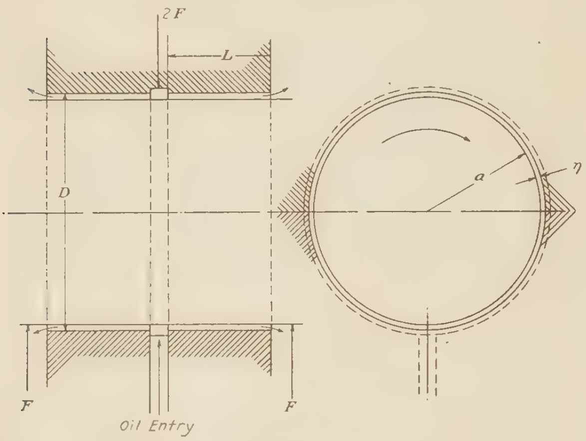

Fig. 1 Diagrammatic Sketch of Bearing Showing Oil-Supply Groove

2 A friction torque per unit length T. For the initial condition, T is the torque required to overcome viscous friction in a uniform layer of oil.

3 A rate of oil flow Q, determined by the initial concentric position of the shaft.

From T and Q, given other necessary data, the mean oil-temperature rise may be deduced. Such an estimate assumes that all friction work corresponding to T is dissipated in the oil.

The journal may then be assumed to take up a succession of positions of increasing eccentricity. Each position is characterized by particular values of the three quantities which together specify bearing performance. Pressure P of course takes finite values for shaft positions other than the initial concentric one, and represents load-carrying capacity. The laws of variation of P, T, and Q are more or less distinct, but they have a common dependence upon eccentricity. Hence, the latter serves as the most convenient independent variable to which the possible operating conditions of the bearing may be referred.

In the following sections, the variation of P, T, and Q with eccentricity will be discussed. It is essential to express each variable in a nondimensional reduced form in order that it may have general application.

Nomenclature for Principal Terms

a = radius of journal, in.

c = eccentricity ratio = $\dfrac{\text{journal eccentricity}}{\text{radial clearance}}$

D = nominal diameter of bearing, in.

f = tangential frictional force per unit area, lb per sq in.

F = load carried by bearing, lb

h = film thickness at any angle θ, in.

h_0 = minimum film thickness, in.

K = end-leakage correction factor for carrying capacity

L = axial length of bearing, in.

m = eccentricity modulus = $1/(1 - c)$

N = rotative speed of journal, rpm

p = oil-film pressure at angle θ, lb per sq in.

p_s = lubricating-oil supply pressure, lb per sq in. gage

P = applied load per unit of projected bearing area, lb per sq in.

P' = reduced expression for bearing pressure = $(P/\mu N)(\eta/a)^2$, nondimensional

Q = volume of oil flowing axially through bearing, cu in. per sec

Q' = reduced expression for oil flow = $Q\mu L/p_s a\eta^3$, nondimensional

S = total friction power of bearing, in-lb per sec

s = friction power per unit bearing surface, in-lb per sq in. per sec

T = friction torque acting on journal, lb-in. per in. of axial length

T' = reduced expression for friction torque = $(T/\mu Na^2)(\eta/a)$ nondimensional

V = rubbing speed of journal and bearing, in. per sec

Z $\begin{cases} = \text{viscosity, centipoises} \\ = \text{(kinematic viscosity, centistokes)} \times \text{(specific gravity)} \end{cases}$

β = angle subtended by effective bearing arc, deg

γ = volumetric heat capacity of oil, in-lb per cu in. per deg F

η = radial clearance, in.

θ = angle measured in the direction of rotation from line of centers of bearing and journal

θ_1 = angle θ measured to point of maximum film pressure

λ = coefficient of journal friction

μ = absolute viscosity, lb-in-sec units = $Z/6{,}900{,}000$

The so-called eccentricity modulus m has been used in preference to eccentricity c for purposes of graphical representation. Both load and friction theoretically approach infinity as c approaches unity. Since these relations are asymptotic, c is an inconvenient variable when eccentricity is high, say above 0.9. By substituting $m = 1/(1 - c)$, an open scale is secured throughout. The variation of both load and friction with m is gradual, and nearly enough linear to facilitate interpolation. Minimum film thickness is also easily expressed in these terms, i.e., $h_0 = \eta/m$.

Bearing Pressure and Eccentricity

One of the familiar expressions in lubrication theory is the nondimensional group here designated as

$$P' = \frac{P}{\mu N}\left(\frac{\eta}{a}\right)^2 = f(c) \quad\ldots\ldots\ldots\ldots\ldots\ [1]$$

The expression is developed from the basic equations for oil-film pressure, which need not be reviewed here. It meets the requirement of a nondimensional group proportional to bearing pressure. As ordinarily used, P' will include a leakage factor appropriate for the particular case; P'_∞ will refer to the corresponding bearing without end leakage.

It has already been remarked that pressure distribution in a complete bearing does not resemble that of the theoretical 360-deg bearing. Load-carrying capacity as a function of eccentricity must be found with reference to conditions approximately as they actually exist. Attempting to determine the relation of P' to m, for bearings of various L/D ratios, data of the following kinds were referred to and utilized in part:

1 Carrying capacities of partial bearings. Capacities of both central and offset types are given by Howarth (1)[2]. Capacities of offset bearings up to $m = 40$ are given by Barber and Davenport (2). Capacities of 120-deg central bearings up to $m = 100$ are given by Needs (3).

2 Leakage factors, to be applied to carrying capacities. Factors suitable to low values of m, and to flat plates under certain conditions, are given by Kingsbury (4). New factors, com-

[2] Numbers in parentheses refer to bibliography at end of paper.

pletely changing the aspect of bearings operating with high eccentricity, have been recently given by Needs (3).

3 Experimental data, showing the extent of the active arc and film pressure distribution in actual complete bearings. Such data are given by Bradford and Grunder (5), and by McKee and McKee (6). Similar data for film-lubricated bearings at extremely high eccentricities are given by Stanton (7).

Finally, friction tests of film-lubricated complete bearings were referred to as a means of choosing between two alternative versions of the load-eccentricity relationship.

Experimental results agree in showing that the actual behavior is that of an offset partial bearing. Therefore, the central-bearing-capacity data were set aside, and attention was confined to selecting the proper method of applying leakage factors to offset-bearing capacities.

In passing, it should be remarked that the series of offset bearings referred to is that described by Howarth (1). The offset positions are determined in accordance with a somewhat arbitrary rule, though a very convenient one. However, these offset positions are not those of maximum capacity for given eccentricity, falling short by several per cent. Whether the offsets actually occurring correspond more closely to maximum capacity, or to the Howarth rule, is not known. At most, the assumption might lead to a slight underrating of bearing capacity.

Two procedures for applying leakage factors were considered. These are outlined briefly as (*A*) and (*B*):

(*A*) This procedure is illustrated in Fig. 2. It is based upon the Kingsbury leakage factors, together with capacity data for partial offset bearings of varying effective arc. Taking any one length/diameter ratio (in this case $L/D = 1$), the proportions of the film-pressure area, and hence the factor K, are determined

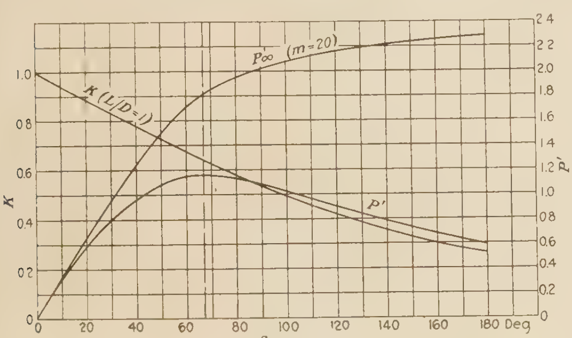

FIG. 2 APPLICATION OF LEAKAGE FACTORS, METHOD *A*
[Factors K from Kingsbury (4); $L/D = 1$; $m = 20$; P'_∞ from Barber and Davenport (2).]

by β. The figure shows the variation of K between $\beta = 0$ and $\beta = 180$ deg.

Likewise, P'_∞ is a function of β; this has been plotted for $m = 20$ ($c = 0.95$). The factor K decreases with β, while P'_∞ increases. Their product $P' \doteq KP'_\infty$ consequently attains a maximum at some intermediate arc; in this case at $\beta = 67$ deg, where $P' = 1.16$. According to this procedure, these figures for β and P' are adopted as representing the probable effective arc and capacity, respectively, for the case considered. Arc β, so determined, was found to vary from a minimum of about 26 deg for a short bearing with high eccentricity to a maximum of about 160 deg under the converse conditions.

The Needs leakage factors were considered unsuited to this purpose, since β varies widely from 120 deg, the arc for which they were experimentally determined.

(*B*) The Needs leakage factors were applied to P'_∞ for 120-deg offset bearings. While the leakage factors were derived for 120-deg central bearings, Needs (3) suggests their suitability for other types. No considerable error was expected in applying the same factors to offset bearings of equal arc. This method assumes that β is always about 120 deg instead of varying widely as in (*A*).

On comparing the respective results with experimental data, method (*B*) was found to agree more closely with fact and was adopted. Active arcs under usual conditions have been found to lie between 90 deg and 150 deg. Further confirmation is found in the friction data, which on the whole fit the results of

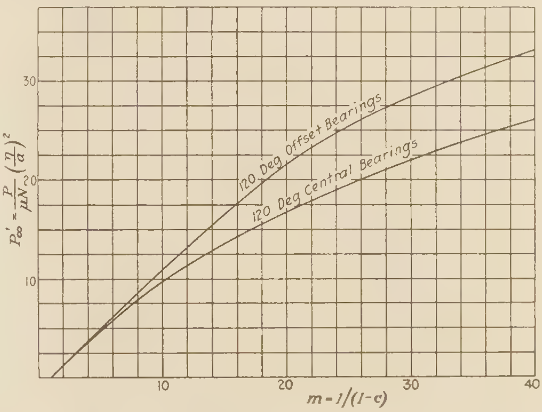

FIG. 3 CARRYING CAPACITIES OF 120-DEG IDEAL CENTRAL AND OFFSET BEARINGS
[Central-bearing data from Needs (3); offset bearing data from Barber and Davenport (2).]

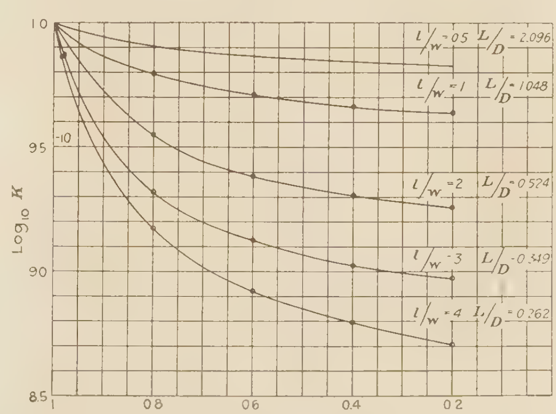

FIG. 4 LEAKAGE FACTORS FOR CARRYING CAPACITY OF 120-DEG PARTIAL BEARINGS AS GIVEN BY NEEDS (3)

(*B*) better than those of (*A*). The reference to method (*A*) has been included because it perhaps represents qualitatively the behavior of bearings under extreme conditions, especially short bearings at high eccentricities. In Stanton's tests (7), where c was 0.9906 to 0.9984, β was 14 deg to 29 deg, approximately, for various cases. The attempt to use the method in calculation is hampered by a lack of suitable load factors K.

Details of the application of leakage factors according to method (*B*) require little comment. Basic capacities P'_∞ for 120-deg offset bearings, plotted against m, are given in Fig. 3. The corresponding curve for central bearings is also shown for comparison, indicating the excess capacity due to the offset assumption as adopted. The difference becomes large only when $m = 10$ or more. Needs (3) quotes values of K varying with c

and with the length-width ratio l/w. To convert these ratios to the terminology used in this paper, note that $w = L$ and $l = \pi D/3$. Hence $L/D = 1.048/(l/w)$. The factor K is shown by Fig. 4, plotted to a logarithmic scale on a base c, to facilitate interpolation at high eccentricities.

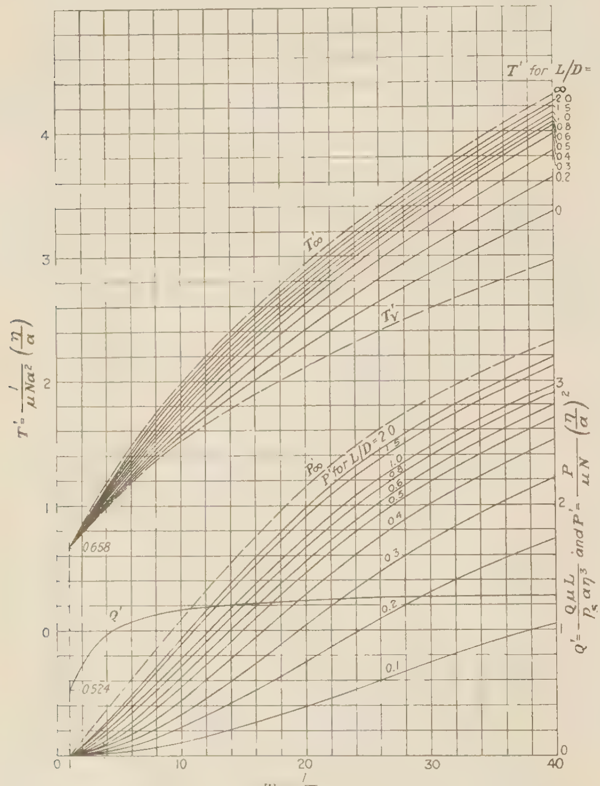

FIG. 5 DESIGN CHART SHOWING RELATIONSHIP OF PRESSURE, FRICTION, AND OIL-FLOW FUNCTIONS TO ECCENTRICITY

The groups of curves at the bottom of Figs. 5 and 5a express the results in terms of P' as a function of m and L/D. They describe the relation of eccentricity to load, in accordance with the assumptions. Friction will now be dealt with in a similar manner.

FRICTION TORQUE AND ECCENTRICITY

It was previously remarked that the friction torque to be overcome in rotating a journal is explicitly related to eccentricity. It is only incidentally related to the applied load, and the coefficient of friction λ is at best a by-product of the analysis and of dubious value.[3] Torque as a function of c (or m) will therefore be calculated consistently with the load computations already made and expressed in nondimensional form. More attention will have to be given to analytical details than in the case of P'.

The elementary equation for tangential frictional force per unit of area on the journal surface is (1)

$$f = \frac{\mu V}{h} + \frac{h}{2a}\frac{dp}{d\theta} \qquad [2]$$

The first term of the right-hand member of Equation [2] arises from viscous friction only. The second term arises from circumferential pressure variations only. For the present case it is desirable to develop the two terms separately. The subscript

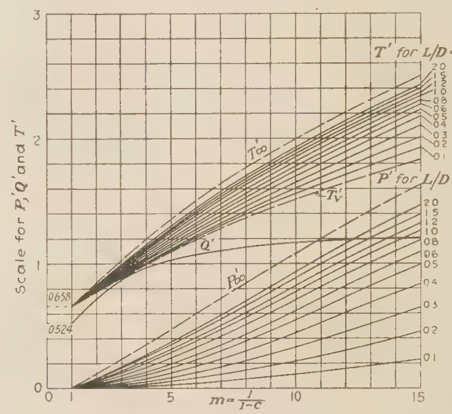

FIG. 5a DESIGN CHART FOR ECCENTRICITIES UP TO $m = 15$

v will refer to the viscous friction and the subscript p to the pressure friction.

Following the assumptions already stated, since the clearance space is completely filled with oil, viscous friction should exist around the whole circumference. But since only 120 deg of arc is active in the supporting load, pressure friction will be confined to this region. Each part of the torque will be calculated separately, and the two then added.

Viscous Friction. Referring to Fig. 6, note that $h = \eta(1 + c\cos\theta)$. Also $V = 2\pi a N/60$ in. per sec. Therefore

$$f_v = \frac{2\pi\mu N a}{60\eta}\,\frac{1}{(1 + c\cos\theta)}$$

Now T_v = viscous-friction torque per unit length and $dT_v = f_v a^2 d\theta$. Therefore

$$dT_v = \frac{2\pi\mu N a^3}{60\eta}\,\frac{d\theta}{(1 + c\cos\theta)}$$

and

$$T_v = \frac{2\pi\mu N a^3}{60\eta}\, 2\int_0^\pi \frac{d\theta}{(1 + c\cos\theta)}$$

which upon integration between limits becomes

$$T_v = \frac{\mu N a^3}{\eta}\,\frac{\pi^2}{15}\,\frac{1}{\sqrt{(1 - c^2)}} \qquad [3]$$

Now introducing the dimensionless counterpart of torque per unit of length, that is

$$T' = \frac{T}{\mu N a^2}\left(\frac{\eta}{a}\right) \qquad [4]$$

[3] The familiar diagram, λ versus ZN/P, is open to criticism on several counts. The reciprocal of bearing pressure is no more serviceable in bearing-load calculations than would be the reciprocal of stress (square inches per pound) in any strength calculation. Its use in graphic representation causes the significant part of the data to be compressed into a narrow corner, to an unreadable scale, while the greater part of the diagram is given over to pressures falling beneath the level of practical usefulness.

The coefficient λ, tending as it does toward infinity at small loads, is likely to obscure the nature of variations in bearing friction. Friction torque passes through three well-defined phases as load is progressively applied: (a) an initial figure for the unloaded bearing, (b) a gradual rise as load is increased, within the limits of film lubrication, and (c) an abrupt rise when incipient metallic contact begins. A rough analogy with the ordinary tensile test of a structural material is noticeable. The three phases in that case become, respectively: (a) unstressed length, (b) elastic extension, and (c) yield.

its value in terms of c becomes

$$T'_v = \frac{\pi^2}{15}\frac{1}{\sqrt{(1-c^2)}} \quad \ldots\ldots\ldots\ldots [5]$$

In order that this amount of friction shall be present, it is only necessary that the clearance be filled with oil. Since this condition is assumed, T'_v will occur unmodified, irrespective of length.

A special case is that of the unloaded journal. Since $c = 0$ $T'_v = \pi^2/15 = 0.6580$. This is the whole amount of friction to be expected, since $dp/d\theta = 0$. Observation of this limiting value of T', either directly or by extrapolation, is a sensitive check upon the accuracy of experimental measurements; yet one apparently little used.

Pressure Friction. Taking the second right-hand term of Equation [2]

$$f_p = \frac{h}{2a}\frac{dp}{d\theta} \quad \ldots\ldots\ldots\ldots [6]$$

The expression for $dp/d\theta$ (1) is

$$\frac{dp}{d\theta} = \frac{6\mu Va}{\eta^2}\left[\frac{c(\cos\theta - \cos\theta_1)}{(1 + c\cos\theta)^3}\right] \quad \ldots\ldots\ldots\ldots [7]$$

Substituting as before to eliminate h and V, and reducing in terms of dT_p

$$dT_p = \frac{\mu Na^3}{\eta}\frac{\pi}{10}\left[\frac{c(\cos\theta - \cos\theta_1)}{(1 + c\cos\theta)^2}\right]d\theta \quad \ldots\ldots\ldots [8]$$

Replacing T_p by T'_p

$$T'_p = \frac{\pi}{10}\int_{\theta_3}^{\theta_4}\frac{c(\cos\theta - \cos\theta_1)}{(1 + c\cos\theta)^2}d\theta \quad \ldots\ldots\ldots [9]$$

This integration may be performed either graphically or analytically. In spite of its lengthy appearance, the analytical expression is preferable especially for high eccentricities. This expression is

$$T'_p = \frac{\pi c}{10}\left[\left(\frac{1 + c\cos\theta_1}{1 - c^2}\right)\left(\frac{\sin\theta_4}{1 + c\cos\theta_4} - \frac{\sin\theta_3}{1 + c\cos\theta_3}\right) - \frac{2(c + \cos\theta_1)}{(1-c^2)^{3/2}}\left(\tan^{-1}\frac{\sqrt{1-c^2}\tan\frac{1}{2}\theta_4}{1+c} - \tan^{-1}\frac{\sqrt{1-c^2}\tan\frac{1}{2}\theta_3}{1+c}\right)\right] \quad \ldots\ldots [10]$$

The angles θ_1, θ_3, and θ_4 in the foregoing expressions are as defined in Fig. 6. The angle θ_1, from the line of centers to the point of maximum pressure is defined for this series of offset bearings (1) by

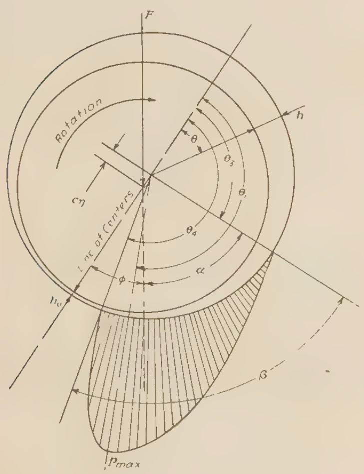

FIG. 6 DIAGRAMMATIC SKETCH OF BEARING SHOWING FILM PRESSURE

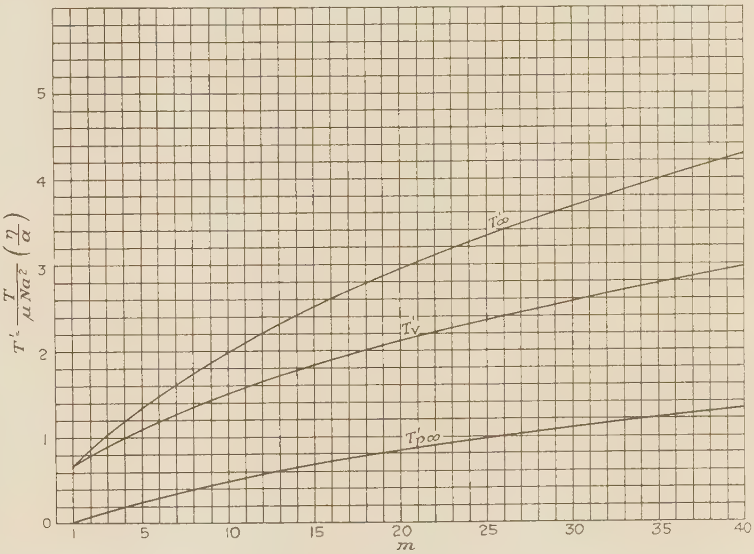

FIG. 7 FRICTION-TORQUE FUNCTIONS WITHOUT LEAKAGE

$$\cos\theta_1 = -\frac{3c}{2 + c^2} \quad \ldots\ldots\ldots\ldots [11]$$

The angles θ_3 and θ_4 define the limits of the active arc; in this case $\theta_4 - \theta_3 = 120$ deg. Equation [10] is to be solved using the same limits, since it is assumed that $dp/d\theta = 0$ outside this region. Further data needed to fix θ_3 and θ_4 include the angle ϕ and the ratio α/β shown in Fig. 6; the information is given in the sources mentioned (1, 2).

Fig. 7 gives the solution of Equation [10] for eccentricities up to $m = 40$. The foregoing derivation takes no account of end leakage; hence, the torque found is that of the infinitely long bearing, and is designated in Fig. 7 as $T'_{p\infty}$. The same figure shows T'_v from Equation [5]. The dominant part played by viscous friction is apparent.

LEAKAGE FACTORS APPLIED TO FRICTION

Viscous friction T'_v is unaffected by end leakage. On the other hand, T'_p is modified by leakage in the same sense that load capacity P' is modified. Hence, the same factors K, already used, can be applied to T'_p with negligible error.[4] This being so, the total friction in a given case becomes

$$T' = T'_v + KT'_p \quad \ldots\ldots\ldots\ldots [12]$$

[4] Strictly, the condition for employing K is that the shape of the circumferential mean-film-pressure curve, with leakage, shall be similar to that without leakage, the ordinates merely being reduced.

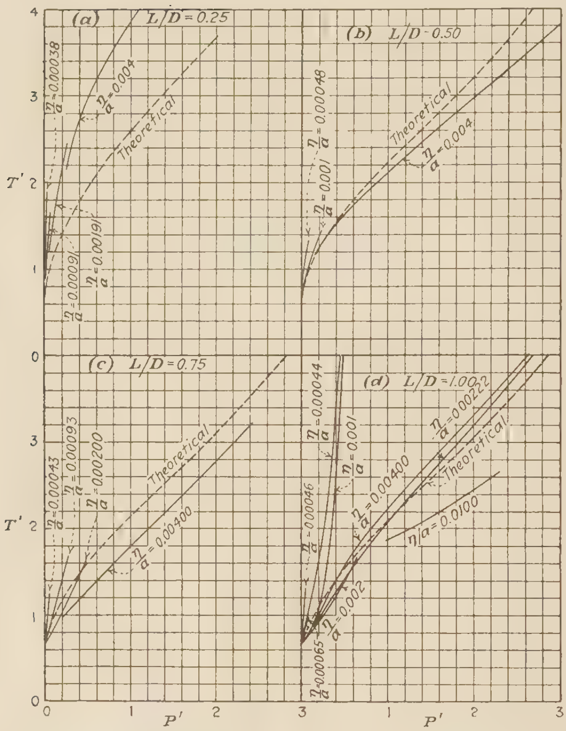

Fig. 8 Experimental Correlation of Load and Friction for Various L/D Ratios
[Data from McKee and McKee (8, 9).]

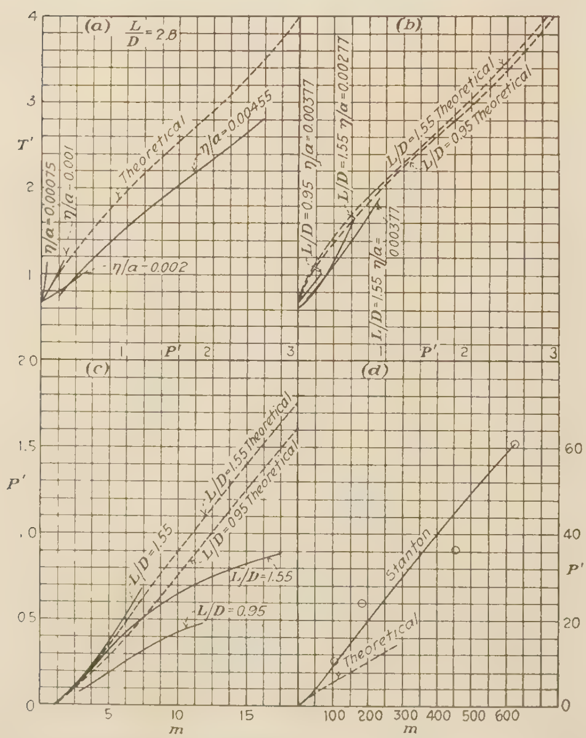

Fig. 9 Experimental Correlation of Load-Friction and Load-Eccentricity
[Fig. a from data by McKee and McKee (8); Figs. b and c from data by Barber and Davenport (2); and Fig. d from data by Stanton (7).]

Figs. 5 and 5a show the values for T' corresponding to the P' curves, for the range covered in each figure.

The diagrams then fully represent the relationships of P', T', and m in accordance with the chosen assumptions. Ordinarily the design data serve to determine P'. Then m and T' may be read at once from the chart. The corresponding physical quantities (running position or film thickness and friction torque) follow by simple calculation.

While a diagram of the type of Fig. 5 is dimensionally in accord with film theory, it is apparent that the exact location of the curves is the outcome of an arbitrary choice of assumptions. Two possible alternatives were discussed early in the paper, and others may suggest themselves. Yet it is probable that the diagram cannot at present be constructed in an entirely satisfactory manner from analytical data. The assumptions adopted, following method (B), are those which lead on the whole to a better agreement with experimental results.

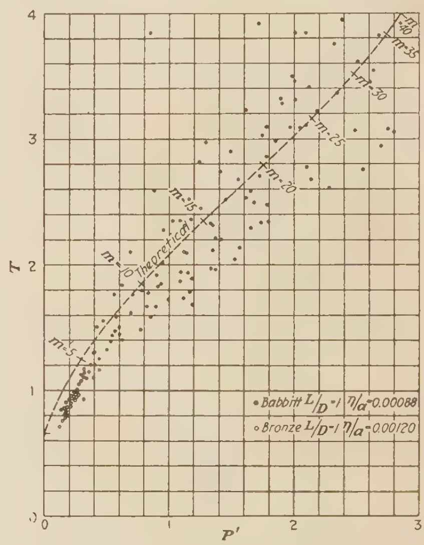

Fig. 10 Experimental Correlation of Thin-Film Load-Friction Results
[Data from McKee and McKee (10).]

The data of Figs. 5 and 5a have been tabulated for reference in Appendix 2.

Correlation of Experimental Data

The greater part of published experimental data for complete bearings relate friction to load, and in addition there are a few figures relating eccentricity to load. Either can be so represented that the extent of departure from the present calculation will be apparent. This has been done for the most pertinent of available data, and the results are given by Figs. 8, 9, and 10. Each test represented is that of a complete bearing, in which the clearance was filled either by supplying oil under slight pressure or by immersion in a bath.

To correlate friction results, P' has been plotted against T', the eccentricity function m being implicit. Fig. 8, Figs. 9a and 9b, and Fig. 10 are of this kind.

Fig. 8 and Fig. 9a represent the results of McKee and McKee for various L/D ratios (8). To Fig. 8d have been added two curves taken from earlier results of McKee, namely, those marked $\eta/a = 0.00222$ and $\eta/a = 0.00044$ (9). These represent bearings tested after extended running-in, while others in this series were not fully run-in at the time of test.

Fig. 9b gives the data of Barber and Davenport for $L/D = 0.95$ and 1.55 (2).

Fig. 10 gives some of the data of McKee and McKee for bearings tested at very low speeds, hence with exceptionally high values of P' (10).

When viewing the P'-T' diagram, it should be recalled that the characteristic behavior of T' is to start from a well-defined initial value, to rise gradually as P' increases, and to jump abruptly where metallic contact begins. In these figures the curves generally have been stopped just short of the critical point where the last sudden rise starts, and as a rule they represent what appears to have been true film lubrication. The reason for thus stopping the curves is that as already mentioned some of the bearings, particularly those of McKee and McKee (8), were not completely run-in before test. The point at which contact begins in a new bearing is no criterion of what may be expected in normal service. Some exceptions should be noted. In Fig. 8*d*, the plot of $\eta/a = 0.001$ is based on tabulated figures in the reference and is extended to show the rapid rise of T' near the critical point. The same is true of the case $\eta/a = 0.00044$ in the same figure. Also, in Fig. 10, all points from Tables 1 and 3 given by McKee and McKee (10) and lying within the scope of the chart have been indicated. The bearings were well run-in before test. Some of the points show by their location the beginning of contact.

Fig. 8*c* represents the data of Barber and Davenport (2) on eccentricity, for two L/D ratios.

Fig. 8*d* gives the results of Stanton (7) under extreme conditions not encountered in ordinary bearing design ($\eta/a = 0.020$ and 0.060). The actual effective arcs were only 14 deg to 29 deg. Eccentricity was as high as $c = 0.9984$ ($m = 625$), yet the load was evidently supported by film pressure only. These cases illustrate the extreme conditions under which true film lubrication may exist. The theoretical curve is merely a rough extrapolation of the corresponding P'-m calculation. This diagram incidentally illustrates the suitability of the present methods in representing a wide variety of conditions.

Referring to the entire group of figures in which a correlation with theory has been attempted, several aspects suggest themselves, which can be discussed separately.

(*a*) *Accuracy of the P'-m Relation.* Fig. 8*c* indicates that at $L/D = 1.55$ agreement is reasonably good, but that at $L/D = 0.95$ the theory makes insufficient allowance for end-leakage effect. Fig. 8*d* indicates that at a still higher L/D ratio $= 2.50$, carrying capacity at a given eccentricity is substantially above the theory. All of these suggest a steeper L/D correction for leakage than has been incorporated in the calculations. The data are meager and scattered, and are exceptionally difficult to obtain. Therefore, though the trend is of interest, it is necessarily inconclusive.

(*b*) *Accuracy of the P'-T' Relation.* A condition which is outstanding in each P'-T' diagram is the discrepancy among the experimental results themselves. On dimensional grounds alone, all tests at a single L/D ratio might be expected to coincide, irrespective of their correlation with theory. In no instance does this happen. The data for $L/D = 1.00$ are the most extensive, and this case is typical. The following curves in Fig. 8*d* are derived from a single series of tests, all made with bearings incompletely run-in: $\eta/a = 0.01000$, 0.00400, 0.00200, 0.00100, 0.00065, and 0.00046. Of these, the first two come nearest to agreement with the calculated curve. The others, with decreasing clearance ratio, fall in positions of progressively steeper slope. On the same diagram, $\eta/a = 0.00222$ and 0.00044 represent bearings differing from the rest only in the fact of having been run-in before testing. These two fall out of their places in the sequence represented by the other six. The former nearly coincides with the calculated curve. The latter is relatively not as steep as $\eta/a = 0.00046$.

In other words, bearings with large clearance ratios and those well run-in come nearest to agreement with theory. Newly machined bearings, and those having exceptionally small clearance ratios, tend to depart from the theory in the direction of increased friction. Both of these trends are explainable in terms of minute roughness of the surfaces. Assuming surfaces of the same quality in two cases where $\eta/a = 0.0100$ and $\eta/a = 0.00100$, respectively, the relative importance of minute projections is greater in the latter instance. Their effect is to cause the bearing to behave as if its clearance ratio were smaller than measured. The two bearings will perform similarly only after the projections are reduced to the point of geometrical similarity, or to such proportions that they no longer play any significant part.

By the same token, surface imperfections will play a smaller part if the journal and bearing are themselves relatively large. Fig. 8*d* is based on tests of a journal $1^1/_4$ in. in diameter, as compared to $2^1/_2$ in. for Fig. 9*b*. Results in the latter case fall somewhat below the calculation. The designer is concerned with relatively large bearings normally run-in. For such cases, a better agreement with theory is to be expected than under some of the exceptional conditions represented in the experiments.

Fig. 10 presents a fair agreement between theory and experiment as far as apparent film lubrication extends. When partial contact begins, T' rises quickly off the scale of the diagram. The most systematic departure from the curve is at small values of P', and amounts to about —20 per cent.

A noticeable feature of many of the tests is a rather close check at the initial point $T'_0 = \pi^2/15 = 0.658$. This agreement bespeaks the accuracy of the data. The principal exceptions are some of the steep curves, for which T'_0 as found by extrapolation is somewhat high.

(*c*) *Accuracy of the L/D Correction.* The foregoing section refers principally to the case $L/D = 1$. The P'-T' relation is somewhat different for other L/D ratios. Inspection of Fig. 8 and Fig. 9*a* gives only a rough idea of the suitability of corrections incorporated in the calculation. Allowing for apparent discrepancies in test data, the various L/D ratios correlate with theory, roughly, as follows: $L/D = 2.8$, indeterminate; $L/D = 0.75$ and 0.50, correlation about equal to that for $L/D = 1$; and $L/D = 0.25$, a wide divergence. The calculations appear to be undercorrected for very small L/D ratios. There is unfortunately little evidence upon which a realignment of this part of the chart might be based.

The principal effect of a reduced L/D ratio is not so much to alter the P'-T' relation as to increase the running eccentricity. Friction tests do not directly afford information in this regard.

(*d*) *Limit of Film Lubrication.* Film thickness estimated from experimental data is often extraordinarily small. Some extreme cases among those represented in the figures will serve as examples. The McKee thin-film data plotted in Fig. 10, indicate that friction was still determined by laws of fluid lubrication about to the limit of that chart, or say to $m = 40$. Radial clearance η for the babbited bearing was 0.00055 in. Accepting the calculated load-eccentricity relation, the film thickness becomes $h_o = 0.00055/40 = 0.000014$ in., before breakdown.

Stanton's calculated film thickness in the tests represented by Fig. 9*d* ranged from 0.000046 in. to 0.000096 in.

The well-run-in bearing, $\eta/a = 0.00222$ as given in Fig. 8*d*, had an apparent film thickness $h_0 < 0.000040$ in. before contact occurred.

These results agree in placing the order of magnitude of the minimum film thickness at 0.00005 in. or less under favorable experimental conditions. It is unlikely that any actual design could successfully be based upon a film thickness of this order. Even assuming the surface extremely smooth, questions of alignment and of foreign particles in the oil remain. Hence, the ex-

periments do not suggest any particular eccentricity or film thickness as being a suitable limit for design purposes.

Conclusions From Experimental Correlation

1 Experimental tests of bearings incompletely run-in are inconsistent and probably misleading. Not only the critical point is affected, but friction is increased throughout the load range. Small bearings with low clearance ratio are most affected while large bearings with relatively large clearance are least affected.

2 Correlation between friction experiments and theory is fair in the case of well-run-in bearings with an L/D ratio of 0.5 or higher. The carrying capacity of very short bearings, in which $L/D = 0.25$ approximately, is probably even less than indicated by Figs. 5 and 5*a*.

3 Meager eccentricity data tend to confirm the conclusion that very short bearings will operate with higher eccentricity than shown by the charts.

4 Oil-film thickness and running eccentricity can be carried to extremes under laboratory conditions which cannot be accepted for design purposes. The experimental results do not suggest suitable limits for use in design.

In the continuation, the relations expressed in Figs. 5 and 5*a* will be accepted as a working basis.

Friction Loss

The design chart, Fig. 5, is entered by way of the dimensionless variable P', which derives from its physical counterpart, an actual bearing pressure P. From the chart is taken T', also dimensionless. It is then a simple matter to determine the actual friction torque and related figures. From Equation [4] by transposition

$$T = \mu N a^2 (a/\eta) T' \qquad [13]$$

where T is expressed in pound-inches per inch of length.

The total torque for the bearing is LT. Friction power expended is

$$S = (2\pi N/60) LT = (\pi/30) \mu N^2 a^2 (a/\eta) LT' \qquad [14]$$

where S is expressed in inch-pounds per second.

Friction power per unit of bearing surface is

$$s = S/2\pi a L = (\mu N^2 a/60)(a/\eta) T' \qquad [15]$$

where s is expressed in inch-pounds per square inch per second. This quantity is significant for high-speed bearings subject to failure by overheating.

The relation between the force F and the friction torque LT which must be exerted to support it is of interest. Referring to Equation [1]

$$F = PLD = 2aL\, P' \mu N (a/\eta)^2 \qquad [16]$$

From Equation [12]

$$LT = LT' \mu N a^2 (a/\eta) \qquad [17]$$

Dividing and simplifying

$$(F/LT) = (2/\eta)\,(P'/T') \qquad [18]$$

The object of a bearing design is to support the load with a minimum exertion of friction torque, hence, this expression in a sense measures the quality of the design. Since it is necessarily dimensional in character, it is not a true efficiency, and in fact no such simple ratio exists. It can be shown that the coefficient of friction $\lambda = 2\,(T'/P')(\eta/a)$, but radius a as well as λ enters into friction-loss calculations.

Curves of P'/T' are plotted in Fig. 11. It is noteworthy that a small L/D ratio is very detrimental in this regard, especially when a minimum film thickness is to be maintained. A gradual improvement in the ratio continues throughout the useful range of m.

The occurrence of η in Equation [18] does not mean that a small clearance in general reduces friction loss, since η also enters into the computation of P'.

Examination of the PV Product

A criterion of the severity of bearing loads which has been used considerably in the past, and is still persistently used in some

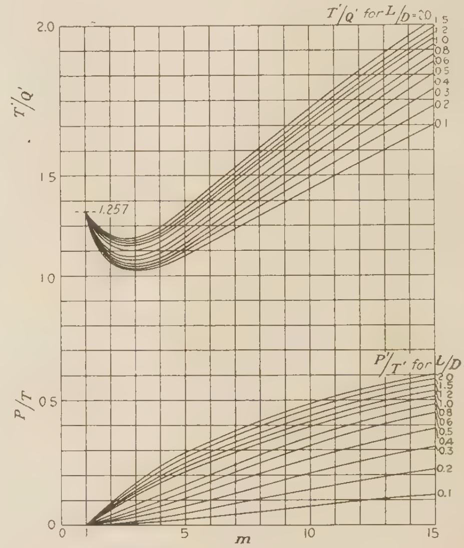

Fig. 11 Functions for Load-Friction and Mean Oil-Temperature Rise

quarters, is the PV product, usually (lb per sq in.) $\times$ (fps). This is, of course, intended as a measure of the amount of friction work per unit of rubbing surface and, were it an accurate measure, would be valuable because of its simplicity. Therefore, a comparison will be made between this friction-work criterion and the corresponding expression based on this analysis.

To form the product PV, note that $V = 2\pi Na/60$ in. per sec and that from Equation [1]

$$P = \mu N (a/\eta)^2 P'$$

Therefore

$$PV = (\pi/30) \mu N^2 a (a/\eta)^2 P' \qquad [19]$$

where PV is expressed in inch-pounds per square inch per second.

The corresponding quantity based on the present calculation is s which is given by Equation [15]. At this point it is not proposed to deal with numerical results derived from either expression, but only with their relative behavior under varying conditions. This is easily done by dividing one into the other. From Equations [15] and [19] and simplifying

$$PV/s = 2\pi (a/\eta)(P'/T') \qquad [20]$$

In order that PV may fairly measure the intensity of loading, this quotient must be constant. The ratio a/η is likely to be about equal in bearings for similar services; therefore, its presence in the expression does not greatly affect the case. A study of Fig. 11 will make it evident, on the other hand, that P'/T' is not even approximately a constant, and that the PV criterion is generally untrustworthy. Aside from its inaccuracy as a measure of friction, it is misleading as a sole test of a design since

it makes no allowance for the great difference in running eccentricity due to variation in the L/D ratio.

Oil Flow Through the Bearing

Axial flow of oil, like load and friction, is a function of journal eccentricity, and the amount of its variation is too great to be ignored. The two limiting cases will serve to illustrate the point. Suppose the journal, while stationary, to be held (a) centrally in the bearing, so that the clearance forms a crevice of uniform width, and (b) in the fully eccentric position, making line contact with its bearing. The flow in the second case will be 2.5 times as great as in the first. When considering the behavior of a bearing lubricated in this or a similar way, it is advisable to take account of the effect, even though the method is approximate. This will be done by ignoring journal rotation (which undoubtedly somewhat retards flow), and also by neglecting the presence of an active bearing arc, where dynamic pressure predominates over supply pressure. The latter omission is of little consequence, since the active arc is generally the zone of least clearance and since some oil is also discharged there as leakage. The assumption of laminar flow is undoubtedly in accord with fact.

The equation for viscous flow between flat parallel plates (11) is

$$p_s = 12\mu Lu/h^2 \qquad [21]$$

in which u = mean oil-flow velocity, in. per sec and p_s = oil supply pressure, lb per sq in. gage.

Substituting $h = \eta(1 + c\cos\theta)$ and transposing

$$u = (1 + c\cos\theta)^2(p_s\eta^2/12\mu L)$$

If Q = oil flow, in. per sec

$$dQ = uhad\theta = (p_s a\eta^3/12\mu L)(1 + c\cos\theta)^3 d\theta$$

and

$$Q = \frac{p_s a\eta^3}{12\mu L}\, 2\int_0^{\pi} (1 + c\cos\theta)^3\, d\theta$$

Integrating between limits

$$Q = \frac{\pi}{6}\,\frac{p_s a\eta^3}{\mu L}\left(1 + \frac{3}{2}c^2\right) \qquad [22]$$

or, grouping nondimensionally

$$Q' = \frac{Q\mu L}{p_s a\eta^3} = \frac{\pi}{6}\left(1 + \frac{3}{2}c^2\right) \qquad [23]$$

The curve of Q' has been added to Figs. 5 and 5a, enabling flow to be estimated in conjunction with other conditions. Referring to Fig. 1, it will be recalled that oil flow, like other quantities, appertains to one-half of the bearing as shown.

Mean Temperature Rise of Oil

Expressions for friction work given by Equation [14] and for oil flow given by Equation [23], may be combined to give the mean oil-temperature rise in passing through the bearing. This estimate assumes that all heat generated by friction is dissipated in the oil, a condition approached in high-speed engines.

A typical lubricating oil at the temperature of bearing service has approximately a specific heat of 0.47 and a specific gravity of 0.915. To fit into the expressions used in this paper, volumetric heat capacity, in mechanical units, is preferred. This capacity is $\gamma = 0.47 \times 0.915 \times 778 \times 12/27.7 = 145$ in-lb per cu in. per deg F.

The heat capacity so expressed varies only gradually with temperature.

From Equations [14] and [23]

$$S = (\pi/30)\mu N^2 a^2(a/\eta)LT'$$
$$= Q\gamma\,\Delta t = Q'(p_s a\eta^3/\mu L)\gamma\,\Delta t$$

Combining and simplifying

$$\Delta t = \frac{0.00072}{p_s}\,\frac{(\mu LNa)^2}{\eta^4}\,\frac{T'}{Q'} \qquad [24]$$

where Δt is expressed in degrees Fahrenheit.

The coefficient 0.00072 corresponds to the case of a complete circumferential groove. Where other means of oil supply are used a new coefficient must be estimated, inversely proportional to flow.

Fig. 11 shows the variation of the quotient T'/Q'. For the concentric shaft, $T'_o/Q_o' = 2\pi/5 = 1.257$. Between $m = 1$

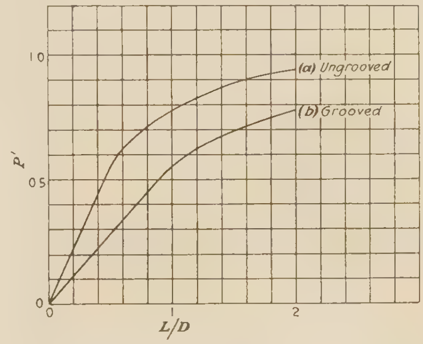

Fig. 12 Effect of Central Groove on Carrying Capacity at Constant Eccentricity, $m = 10$

[The ratio L/D is based on total length in each case. The width of the groove has been neglected in plotting curve b.]

and $m = 6$, approximately, the ratio drops below its initial value. In this region, increase of flow due to eccentricity is greater than increase of friction. While Equation [24] is not to be taken as accurate, in view of the assumptions involved, it is a serviceable means of estimating for design purposes. The adequacy of oil supply is measured by temperature rise so expressed.

The viscosity μ wherever it occurs can be taken as the mean of its values at entry and at exit. The in-going viscosity being known, there is no direct way to solve for the average since the viscosity-temperature relation is not easily expressed functionally. Equation [24] enables an average to be found by trial and error. No experimental results are available for checking Equation [24] nor the expression for oil flow, Equation [22].

Alternative Methods of Oil Feed

The preceding discussion has assumed that oil is supplied through a groove running circumferentially around the bearing at a point midway of its length. From the cooling standpoint, this is unquestionably the best arrangement that can be made. Such bearings are widely used in high-speed engines, and their durability under severe loads can only be explained as due to evenly distributed forced cooling. The weakness of the design is that the effective L/D ratio is cut to about one half, as compared to the same bearing without a groove. Fig. 5 shows by inspection that the grooved bearing will in consequence run with a much higher degree of eccentricity than one in which the pressure surface is unbroken. Success of the grooved-type bearing emphasizes the importance of forced cooling, where rubbing speeds are say 20 fps or higher.

The loss in carrying capacity due to a central groove is shown by Fig. 12. Both curves are drawn for the same eccentricity,

$m = 10$. Curve (*a*) is for a bearing of length L in which the pressure surface is unbroken while curve (*b*) is for one of equal overall length, but divided by the central groove. The width of the groove itself has been neglected. Loss of capacity is greatest for cases in which the overall length is already small, and is substantial in any case. In so far as it may be done without sacrifice of effective cooling, the groove should be avoided.

Oil Holes. Oil-supply holes drilled in either bearing or journal, midway of the length, cause a forced flow across the bearing face very similar to that from a groove. The attendant loss of carrying capacity is not so great, and can be minimized by locating the holes outside of regions where the load diagram shows that high pressures are to be expected.

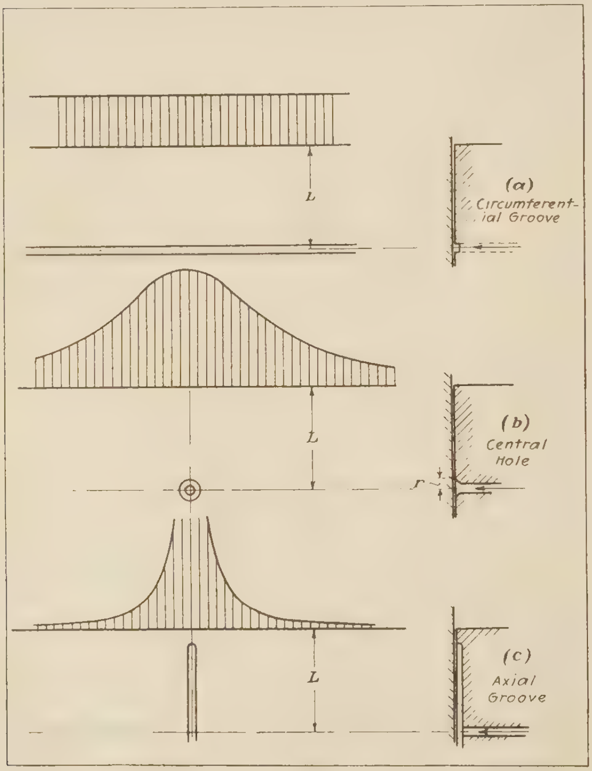

Fig. 13 Alternative Pressure Lubrication Schemes Showing Distribution of Oil Flow

The preceding discussion of oil flow assumed a continuous groove. Where holes are substituted, the capacity for flow, and hence for cooling, is usually reduced. It is desirable to compare the two systems in a roughly quantitative way, and this has been done in Appendix 1 with the following results.

Let L equal the half-length of the bearing, and assume an opening of radius r on the center line as shown in Fig. 13*b*. Note that this radius is that at which the rounded opening becomes tangent with the surface. Flow from such an opening can be expressed in terms of an equivalent length x of the circumferential groove, about as follows:

r/L	x/L
0.025	0.68
0.050	0.82
0.100	1.04
0.150	1.23

The circumference being πD, the number of holes equivalent to a complete groove is $\pi D/x$. It is usually not possible to use this number, at least not without defeating the original purpose. The restriction can be offset by an increase of clearance, but some loss of cooling is unavoidable.

Axial Grooves. Oil grooves cut axially and extending along the bearing to a point near each end as shown in Fig. 13*c* appear to serve no purpose that cannot be better served without them. The same is true of diagonal grooves and of recesses or pockets at the parting line. It is sometimes argued that an axial groove when properly located supplies oil where it will be swept into the region of high pressure. This viewpoint overlooks the fact that in a pressure-fed bearing the clearance is continuously filled with oil and a surplus is being discharged at the ends. An amount in excess of that needed to fill the clearance cannot play a part in forming the load-supporting film and is merely superfluous. Hence, there is a loss of carrying capacity at one point of the circumference, without a compensating gain.

With regard to cooling, the axial groove is ineffective. The flowing oil takes the easiest path, which is through the groove to its end, rather than across the bearing face. Therefore, relatively little oil is forced to take a route such that it will efficiently remove heat from the metal surfaces.

The type of flow to be expected with each oil-feed system is indicated in Fig. 13.

Application of Design Charts

The charts may be applied to any case for which the usual design figures are known. It is customary to calculate pressure at crank-angle positions throughout the cycle. When the running speed, clearance ratio, and oil viscosity at the estimated operating temperature are given, the corresponding P', m, and T' can be found.

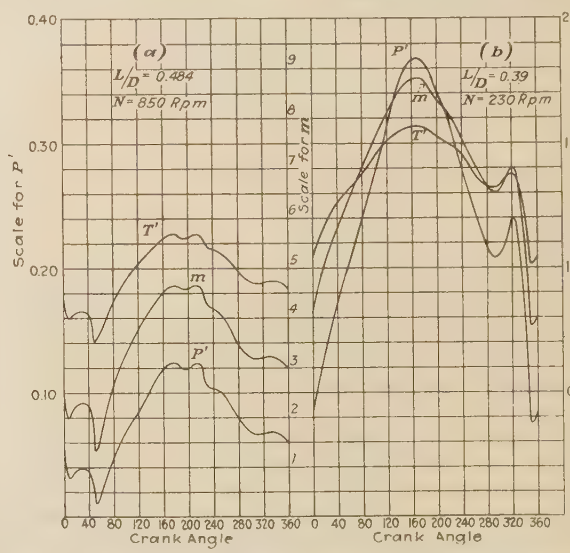

Fig. 14 Analysis of Two Diesel-Engine Center Main Bearings

Fig. 14 illustrates this application in two cases. Each represents the center main bearing of a four-cycle engine, the load cycle being completed in a 360-deg crank angle. Fig. 14*a* is taken from a high-speed engine while Fig. 14*b* is taken from a low-speed engine. In some respects these cases are typical. On inspection (*a*) appears to be less severely loaded; P', m, and (to a less degree) T', are all lower. The great difference in the two conditions is in the friction power generated per unit of surface, that is, in s as given by Equation [15]. Some of the pertinent data for each case are given by Table 1. The mean value of T' is obtained with a planimeter from Fig. 14.

TABLE 1 ENGINE MAIN-BEARING DATA. (FIG. 14)

Case	(a)	(b)
Running speed, rpm	850	230
Shaft radius a, in.	3.484	5.000
Radial clearance η, in.	0.00294	0.0050
Oil viscosity Z, centipoises	60	47
Oil viscosity μ, absolute	9.2×10^{-6}	6.8×10^{-6}
Bearing effective length L, in.	3.375	4.09
L/D	0.484	0.390
Eccentricity modulus (max) m,	4.7	8.8
Film thickness (min) h_0, in.	0.00063	0.00057
h_0/a	0.000181	0.000114
Friction power per unit surface s in-lb per sq in. per sec	447	41

With reference to Table 1, specific friction power s is more than ten times greater in case (a) than in case (b). Bearing (b) is in practice built with a tin-base babbitt lining, and has a long record of trouble-free service. Bearing (a) is lined with a high-lead composition. It has a much shorter history than (b). While it appears to be successful, it gives evidence of working under severe conditions, and a shorter service life is to be expected.

The described procedure lends itself to a systematic investigation of variations in bearing dimensions, clearance ratio, speed, viscosity changes due to temperature, etc.

When applied to a crankpin bearing, allowance should be made for the influence of crank-connecting-rod ratio r/l upon relative angular velocity. If the normal angular velocity be taken as unity, the maximum is $(1 + r/l)$ at top center while the minimum is $(1 - r/l)$ at bottom center.

Criteria for Safe Design

Three quantities suggested by this study jointly define the character of operating conditions in a given case. Each of these should be held within a safe limit to insure satisfactory operation and service life. These quantities are (a) the friction power per unit of surface s as given by Equation [15], (b) the ratio of minimum film thickness to shaft radius, $h_o/a = \eta/am$, and (c) the mean rise of oil temperature Δt as given by Equation [24].

Quantity (a) defines the severity of service from a standpoint of heating alone, a frequent cause of failure in bearings operating at high speed. It is proposed to check this aspect of design by s in place of PV. The permissible upper limit of s differs according to the type of bearing metal employed.

Quantity (b) corresponds to the liability to wear, scuffing, etc., due to near approach of the surfaces. The ratio h_0/a is selected in preference to h_0 because potential failure arises from misalignment, ovality, etc. (defects which generally are proportional to the size of bearing) and not from inability of a very thin film to function in accord with theory. It was shown that in so far as the film itself is concerned, under experimental conditions, h_0 may be far smaller than could be permitted in practice. Therefore, a suitable lower limit can be based upon service experience only. When a bearing is to be designed for operation at widely varying speeus, h_0/a is more likely to be fixed by the low-speed condition than by the high-speed condition.

Quantity (c) determines the adequacy of oil circulation. An upper limit should be set, independent of materials or other conditions.

In order to establish a numerical figure for each of these limits, to be suitable for general application, it would be necessary to review a wide variety of cases. The examples to be considered should include successful, unsuccessful, and border-line types. There has been no opportunity for such an investigation, and no figures can be quoted with confidence. This is especially true of the heating function s, since metals commonly used differ widely in their capacities. For h_0/a, a low limit of 0.00010 is suggested, and for Δt, a value of not over 50 F.

When a design is undertaken which departs in some respects from previous practice, there are usually at hand sufficient data on bearings already known to be successful in the kind of service contemplated. Should it be possible to adhere in the proposed type to a set of limits defined by calculation of known cases, equally satisfactory results can be expected.

Conclusion

The described procedure is believed to be a satisfactory approximate method of applying the data of lubrication theory to engine-bearing design.

The diagrams shown in Figs. 5 and 5a, expressing the results of calculation, are in some degree confirmed by friction experiments. The attempt at correlation is hindered by what appear to be inconsistencies in the experimental data. The least satisfactory check is at low L/D ratios, for which the actual carrying capacity is probably less than indicated by the charts. Further experimental and analytical data should permit a more accurate detailed construction of these charts.

Three criteria are proposed which define the operating conditions of any bearing installation. Insufficient data are at hand to permit dependable numerical limits to be quoted. They may be established for a particular type of service on the basis of known satisfactory examples.

In closing, the author wishes to acknowledge his indebtedness to A. D. Andriola of the Electric Boat Company, Groton, Conn., who completed a large proportion of the numerical calculations, and who also contributed the expression for T'_p as given by Equation [10]. He also wishes to express his appreciation of the interest shown by E. Nibbs, Chief Engineer of the Electric Boat Company.

Appendix 1

Oil Flow From a Central Hole

With reference to Fig. 15, the relative rates of flow from a circular hole of effective radius r and from a continuous groove, respectively, are found by considering two flat parallel plates, the circumferential variation of h being ignored.

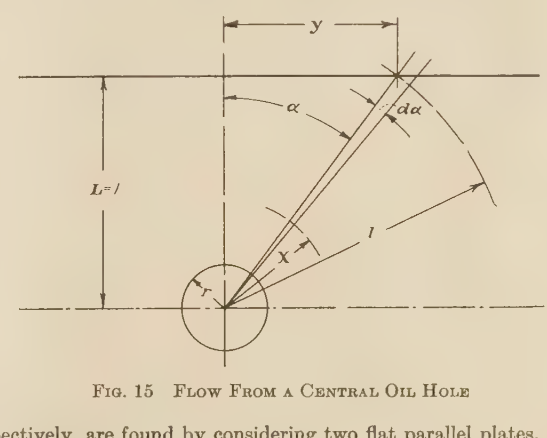

Fig. 15 Flow From a Central Oil Hole

Actual paths of flow will follow a rather complex series of curves. As a rough approximation, this curvature is neglected, and oil is assumed to flow radially in straight lines to the edge of the bearing. The half-length L is taken equal to unity.

Let dQ be an element of volume flowing along the path $d\alpha$. From Equation [21], taking the distance x along the path of flow

$$\frac{dp}{dx} = \frac{12\mu}{h^2} u$$

Now $u = dQ/dA$, in which A = flow section and $dA = hxd\alpha$. Substituting

$$\frac{dp}{dx} = \left(\frac{12\mu}{h^3}\right)\frac{dQ}{d\alpha}\frac{1}{x}$$

By integration

$$p_s = \frac{12\mu}{h^3}\frac{dQ}{d\alpha}\log_e\frac{l}{r}$$

in which l = length of path at angle α. Then if

$$p_s h^3/12\mu = B$$

$$dQ = BD\alpha/(\log_e l/r)$$

Since $l = \sqrt{(y^2 + 1)}$, $\alpha = \tan^{-1}y$, and $d\alpha = dy/(y^2 + 1)$

$$dQ = \frac{Bdy}{(y^2+1)\log_e\left(\sqrt{(y^2+1)}/r\right)} \quad \ldots\ldots [25]$$

The volume of oil Q is found by graphical integration of Equation [25].

For a continuous groove, neglecting the width of the groove itself, and since $L = 1$, the flow per unit of peripheral length is

$$Q = p_s h^3/12\mu = B$$

The desired result is the number of units of peripheral length of groove equivalent to an opening of radius r, which is given by Q/Q_1, or can be found by eliminating B from Equation [25].

Appendix 2

TABLE 2 DATA OF FIGS. 5 AND 5*a*

M		1	5	10	15	20	25	30	35	40
C		0	0.800	0.900	0.933	0.950	0.960	0.967	0.971	0.975
	L/D									
P'	0.1	0	0.023	0.114	0.230	0.40	0.56	0.73	0.91	1.04
P'	0.2	0	0.053	0.228	0.455	0.76	1.03	1.29	1.54	1.72
P'	0.3	0	0.086	0.338	0.66	1.05	1.40	1.71	1.99	2.21
P'	0.4	0	0.121	0.450	0.85	1.30	1.68	2.02	2.28	2.52
P'	0.5	0	0.161	0.551	1.00	1.47	1.86	2.21	2.46	2.71
P'	0.6	0	0.195	0.625	1.10	1.56	1.96	2.29	2.56	2.80
P'	0.8	0	0.251	0.712	1.20	1.67	2.07	2.40	2.65	2.89
P'	1.0	0	0.293	0.778	1.27	1.76	2.14	2.48	2.73	2.96
P'	1.2	0	0.320	0.825	1.33	1.82	2.22	2.54	2.80	3.03
P'	1.5	0	0.349	0.892	1.40	1.90	2.30	2.63	2.88	3.11
P'	2.0	0	0.380	0.940	1.47	1.98	2.38	2.70	2.95	3.19
P'	∞	0	0.484	1.084	1.638	2.154	2.540	2.844	3.096	3.32
T'_v		0.658	1.097	1.510	1.834	2.108	2.350	2.570	2.772	2.960
$T'_{p\infty}$		0	0.258	0.488	0.677	0.840	0.978	1.103	1.218	1.325
	L/D									
T'	0.1	0.658	1.110	1.562	1.935	2.27	2.57	2.86	3.11	3.35
T'	0.2	0.658	1.125	1.613	2.030	2.41	2.75	3.08	3.36	3.63
T'	0.3	0.658	1.142	1.662	2.11	2.53	2.89	3.24	3.54	3.83
T'	0.4	0.658	1.160	1.710	2.19	2.62	2.99	3.35	3.67	3.97
T'	0.5	0.658	1.181	1.754	2.24	2.68	3.06	3.42	3.73	4.04
T'	0.6	0.658	1.203	1.786	2.28	2.72	3.10	3.46	3.77	4.08
T'	0.8	0.658	1.232	1.829	2.33	2.77	3.14	3.50	3.82	4.11
T'	1.0	0.658	1.252	1.858	2.36	2.80	3.18	3.53	3.85	4.14
T'	1.2	0.658	1.267	1.881	2.38	2.82	3.20	3.55	3.87	4.16
T'	1.5	0.658	1.284	1.904	2.41	2.85	3.23	3.58	3.90	4.19
T'	2.0	0.658	1.301	1.931	2.44	2.88	3.27	3.62	3.93	4.23
T'	∞	0.658	1.355	1.998	2.511	2.948	3.328	3.673	3.990	4.285
Q'		0.524	1.028	1.160	1.210	1.232	1.249	1.260	1.263	1.270

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Locomotive and Car Journal Lubrication

By E. S. PEARCE,[1] INDIANAPOLIS, IND.

This paper discusses the economic significance of railroad-journal lubrication and reports on researches made on the causes of failure and development of remedies therefor as pertain to design, materials, and operating practices.

THE subject of railroad lubrication should not be viewed in the light of locomotive miles per pint of valve oil or car oil, or per pound of grease, or car miles per pint of oil, or even these figures equated to dollars and cents and expressed in cost of lubrication per thousand miles.

Such statistics and the practice of measuring efficiency on this basis obscure entirely the economic purpose of lubrication. The object of lubrication is to insure continuity and dependability of operation of the various units of railroad rolling stock, and reduce friction and consequent wear resulting from operation. Lubrication costs as such are incidental. They are the premium on insurance against the effects of wear.

Railroad rolling-stock lubrication is divided into two general classes:

(1) Car lubrication, which may be subdivided into lubrication of freight cars and lubrication of passenger cars. All journal lubrication of this class uses oil and waste.

(2) Locomotive lubrication, which may be divided as follows:

- (*a*) Lubrication of steam cylinders and valves of the locomotive, booster, air pump, and feedwater heater in which oil fed by some mechanical means is used
- (*b*) Lubrication of machinery of valve motion, guides, and crossheads in which oil fed by some mechanical means is used
- (*c*) Lubrication of driving journals and main and side-rod bearings, in which hard grease is used and the heat of the bearing makes it possible to feed the grease in a liquid state. This is also journal lubrication.
- (*d*) Lubrication of engine and trailer trucks and tender journals, in which oil and waste are used in the same manner as with cars, and
- (*e*) A type fast growing in importance, which may be classified under the general term of automotive lubrication such as lubrication of internal-combustion engines with oil.

These classes may be regrouped into two, viz., the lubrication of journals, and the lubrication of pistons, cylinders, valves, and connected parts. One may be considered the lubrication of load-carrying parts and the other the lubrication of power-generating parts.

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Contributed by the Railroad Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held in New York, N. Y., December 2 to 6, 1935.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1936, for publication at a later date.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Journal lubrication has nothing in common with other classes of railroad lubrication and has little in common with industrial practice, for which reason it has received little consideration from the standpoint of research and development. It is not contended that everything is known on the subject of lubrication of valves, cylinders, and pistons of steam engines and internal-combustion engines, but considerably more is known of this form than of journal lubrication. To the general class of journal lubrication this paper is primarily addressed.

From the publication *Railroad Facts*, 1934 edition, published by the Western Railroads Committee on Public Relations, considering the statistics for 1926 as normal, we find that the class-1 railroads make annually 3,800,000,000 passenger-car miles and 28,600,000,000 freight-car miles. This service is performed with 62,000 locomotives, 2,348,000 freight-cars, and 54,000 passenger-train cars. Roughly, therefore, journal lubrication involves the performance of some 20,500,000 journals.

Each journal has a bearing, and all but a relatively small percentage are produced by bronze-casting manufacturers. The following, is taken from a report of the Copper and Brass Research Association, March 16, 1931, entitled—"A Survey of the Bearing and Bushing Industry."

> From data which we have obtained, it is evident that the annual turnover of copper in the bearing market is in excess of 300,000,000 pounds. More than two-thirds of the total is accounted for by the railroad industry, a market whose extent can be figured with a fair degree of accuracy.

This publication then estimates that there are in service on railroad cars and locomotives 370.000 tons of bearing bronze of which 260,000 tons, or 70 per cent, are journal bearings on cars and locomotives.

The annual turnover of this 260,000 tons amounts to 70,500 tons or 27 per cent. Of this total 56,255 tons or 80 per cent are freight-car bearings. On a basis of $5\frac{1}{2} \times 10$ journal bearings weighing 25 lb, this means the removal and application of 4,100,000 journal bearings annually, or about two per car per year.

To check this figure a survey was made for a period of three years on two class-1 Eastern railroads, and it was found that, on the basis of 10,000 freight-car miles, one railroad used from 2.6 to 2.96 bearings per car and the other from 1.97 to 2.54. Passenger cars on one railroad on the same basis of 10,000 miles used from 1.28 to 1.4 bearings per car and another from 0.99 to 1.43. But in this connection it should be recognized that a passenger car makes in one year's time from eight to ten times the mileage of a freight car and, therefore, on a yearly basis the number of passenger-car bearings consumed per car was considerably greater than that of freight-car bearings.

During the same period, on these two railroads, on the basis of 10,000 freight-car-miles, from 1.8 to 3.25 gal of car oil was used, and on the basis of 10,000 passenger-car miles, from 1.54 to 3.35 gal. It is interesting to note that more oil did not mean fewer bearings.

The degree to which lubrication failures contribute to the number of wheel changes on a railroad presents two interesting conclusions. On one railroad, where 12,000 pairs of wheels were removed per year in running repairs, it was found that from 20 to 40 per cent, depending on the season, were removed for cut journals only, the remaining 60 to 80 per cent for other defects, practically all those of the wheel. This illustrates the percentage by which a considerable item of indirect expense would be reduced

by improved lubrication. It emphasizes the fact that any mechanical innovations conducive to improved journal performance must not complicate or decrease the facility with which wheels can be changed.

To arrive at the relative operating significance of lubricating costs there are two elements to consider:

(1) Direct cost of lubricating materials and the labor of applying and handling.

(2) Indirect cost due to wear and replacement of parts; failure of units due to inoperative conditions of bearings; excessive use of power due to unnecessary friction; and the resultant extent to which the deficiencies of lubrication, due to the accumulative effect of the foregoing conditions, become a limiting factor in the operation of the units of rolling stock on the railroads.

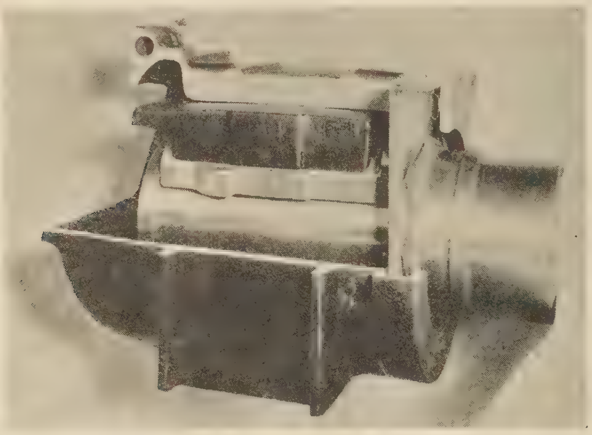

Fig. 1

As all railroad costs are predicated on I.C.C. accounting rules and regulations, it is unfortunate that specific cost data are not available on these items. It would, therefore, be necessary to make specific studies, but it is safe to say that direct lubricating costs alone are in excess of $15,000,000 per year. To this must be added the cost of bearings and all the other items contained in the indirect cost, which to one acquainted with the cost of locomotive and car repairs is easily recognized as a figure several times that of the direct cost of lubrication.

The present journal bearing box is shown in Figs. 1 and 4. Its elements are the axle, journal, bearing, wedge, waste, oil, dust guard, lid, and box, each of which has its effect on performance of the unit as a whole and upon the maintenance of each other. Practically no two elements are the products of one manufacture or class of manufacture. The quality or development of one feature may be nullified by the deficiency of one or more of the others, and all by lack of proper maintenance standards or practice of the owner or user.

The actual situation in regard to the railroad journal bearing was summed up in a discussion of a paper on heavy-duty antifriction bearings before The American Society of Mechanical Engineers, December, 1928. While it is not represented that the quotation was directed specifically to the bearings that are the subject of this paper, nevertheless it is extremely applicable. The discusser said:

> I admit frankly, from a contact of 22 years with the antifriction-bearing business, that if the plain bearing had been developed by specialists and always kept in the hands of its friends, our problem might not have been so easy as it has been.
>
> That, I think, is an excellent example of the axiom that what is everybody's business is nobody's business. Every one has assumed that he knew everything that was to be known about a plain bearing, and the result has been that there have been too many wrong principles in the design, manufacture, and application of plain bearings.
>
> The antifriction bearing has been developed by its friends, and through that concentration of effort the sum total of all the information which has been available has been disseminated, I think usually wisely, for the benefit of the user.

It may be assumed that the present journal bearing and its related parts must have given a great measure of satisfaction in the past because it has remained practically unchanged in the face of advances in mechanical development in the last twenty years.

The present demand for higher speeds, in some cases with heavier loads, and ever-increasing operating periods, has intensified the demand that journal performance be improved or become increasingly a factor in retarding economic railroad operation, thus further nullifying the money and effort already invested in improvements in equipment, materials, and facilities.

The railroad journal bearing is of simple mechanical construction. To date little is known of the fundamentals upon which its positive successful operation depends. The first fact in a constructive analysis for the purpose of finding a starting point for improvement is that there is no base line from which to measure the performance so far obtained nor to measure the degree to which performance can be improved, nor the economics of such improvements as are developed. It is further evident that guesswork has been all too prevalent in the past and is too expensive for the future.

Journal Testing Plant

As little technical information is available, the first step in any advancement of the art is to provide adequate means of observing and collecting data representative of operating conditions. It is also necessary to have means of measuring the nature and the degree to which various elements of bearing construction, lubrication, and operation affect the performance as a whole and the performance of each other.

In order to collect bearing data to provide means by which this improvement can be charted and its economic value measured, the journal testing plant here described was put in operation some five years ago and has been in continuous operation since that time.

While it seemed at the time the building of this plant was under consideration there was ample justification for such a step, the results so far achieved made the original considerations seem of insignificant consequence.

Figs. 2, 3, and 4 show the front, rear, and right side, respectively of the main testing unit. Fig. 5 shows the control-room equipment.

The plant consists of a unit using a standard A.A.R. $5^1/_2 \times 10$-in. axle, journal, brass, wedge, and box, the standard construction in general use on one end of a standard A.A.R. axle. This axle is supported by two large roller bearings that are flood-lubricated by individual pumps for each bearing, circulating a stream of oil through the bearing and a cooling system in order that conditions may be held constant. At the end of the axle opposite the test-journal assembly is a calibrated motor of ample size to start under any condition of load up to 30,000 lb per journal and operate under this or any other load at any speed and at any rate of acceleration. The load on the journal is applied to the top of the journal box in the conventional manner as in actual service, through a coil spring and equalizer bar connected to a suitable lever arrangement, in which the imposed load is balanced on the platform of an indicating scale. In this manner the load can be varied to any degree at any time and is accurately measured.

Accurate calibration of the plant has established correction

factors for all conditions of load, speed, and temperature. By virtue of the calibrated motor driving the axle there is recorded in the control room directly the reading as to the power consumed. Indexed with this chart is a record also of speed. Temperature readings are taken at three points within the bearing, and by a suitable thermocouple arrangement the film temperature between the journal and bearing surfaces is taken also at three points, as well as the temperature of the journal packing in the box beneath the journal. Duplicate readings are taken on standard instruments.

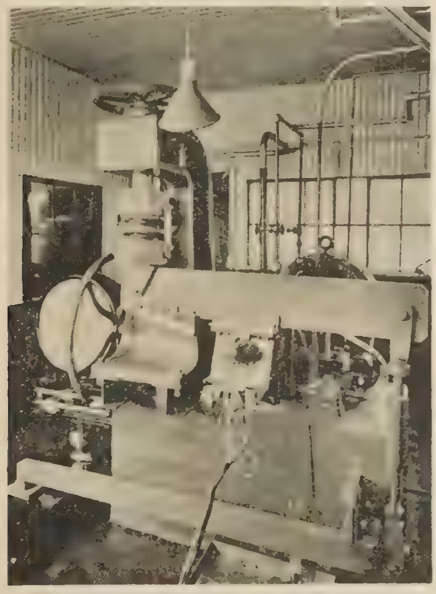

Fig. 2

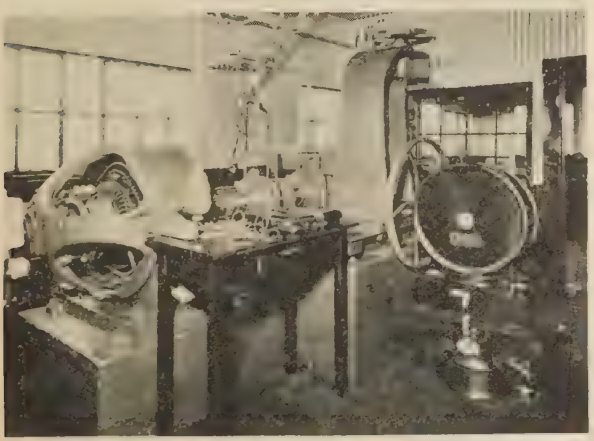

Fig. 3

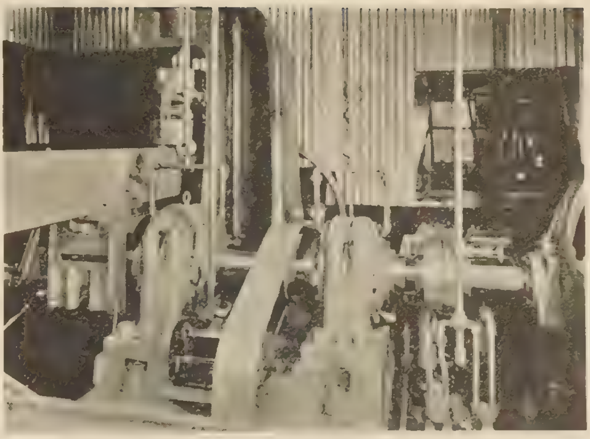

Fig. 4

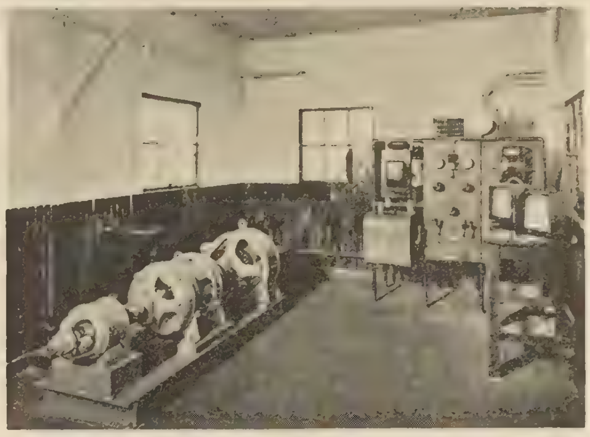

Fig. 5

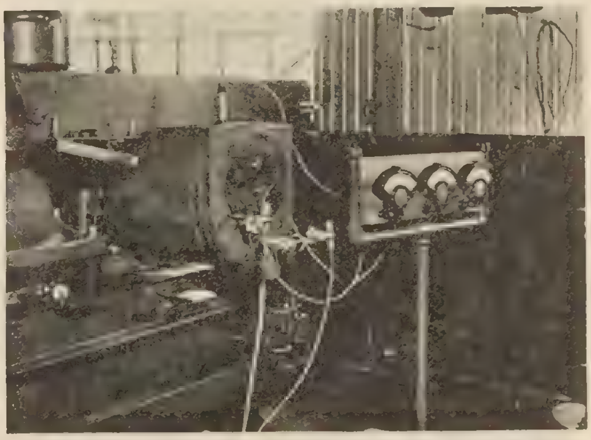

Fig. 6

Openings are cut in the box in such a manner that observation of the journal in its relation to the bearing is possible at all times. In order that journal operation may be observed at low temperatures, a standard $5^1/_2 \times 10$-in. journal box is surrounded by a suitable jacket connected to a refrigerating unit so that the temperature of the box may be lowered as desired. Fig. 6 shows the method of operation under this condition.

The equipment was built to take the standard $5^1/_2 \times 10$-in. journal and journal box assembly instead of a 5×9-in. or 6×11-in. because a survey showed that this size represented the one used on the majority of equipment in this country. Another reason was to eliminate the criticism that has heretofore existed when the mechanical effect of friction has been determined on small-scale equipment and the results equated to that of actual practice.

With this equipment the reliability of any specific element of journal operation can be established. Any single factor affecting the operation such as speed, load, temperature, kind of oil, kind of waste, and construction or composition of bearing can be varied at a time and all others held constant. When tests involving change of speed alone are run, an eight-hour run is made at each speed, usually 5, 10, 30, 50, and 70 mph. The machine is operated continuously during this period for eight hours at each speed; complete observations are made of all temperatures. Charts of speed and power consumed are also made.

Fig. 7

Fig. 8

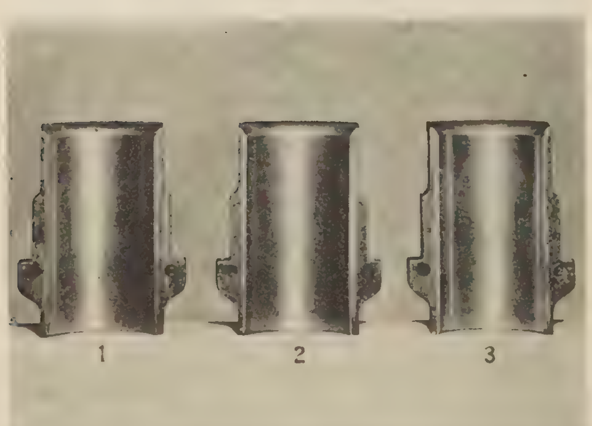

Fig. 9

Fig. 10

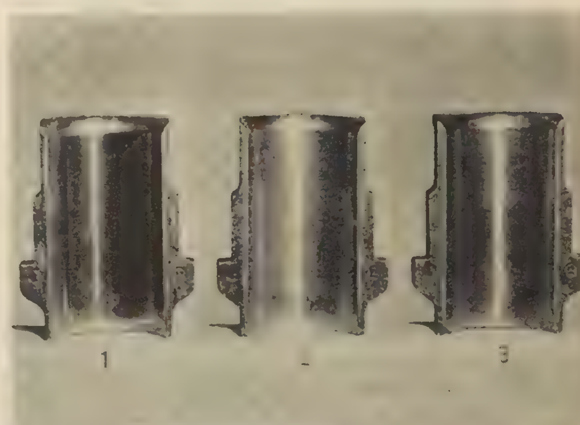

Fig. 11

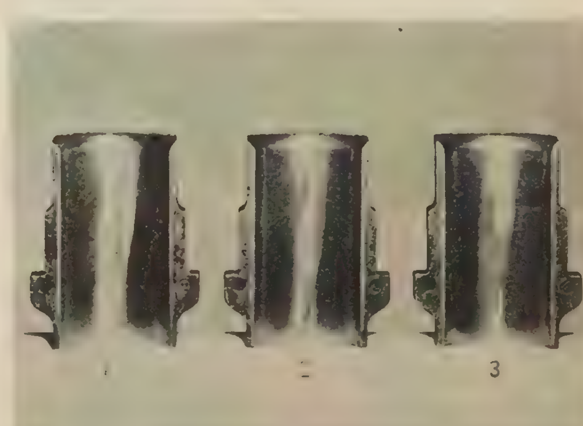

Fig. 12

Bearings

In railroad practice it has been a long-disputed question whether bearings should be broached before application or applied to the journal rough and unbroached. The first few hours of operation with any plain bearing are the most critical.

The usual industrial practice after the bearing has been properly broached, scraped, and fitted is to restrict the speed and load to which the bearing is subjected until the bearing has had an opportunity to wear in. The railroad journal bearing, and particularly the car journal bearing, cannot be subjected to a controlled condition of restricted speed and load because of the peculiar conditions of railroad operation. Immediately on its application it must take its place with and deliver a performance equal to the other bearings under whatever condition of load and speed may be required. Therefore, the railroad bearing should be adequately prepared since its satisfactory operation in severe service is of first importance.

The practice of broaching car journal bearings by the manufacturer or the user has been quite general. However, the economy and justification for this operation have been open to question, in view of the fact that many bearings so prepared fail during the initial service period. In some cases this has led to abandoning broaching and applying rough bearings directly to the journal. Unfortunately, a distinction has never been made

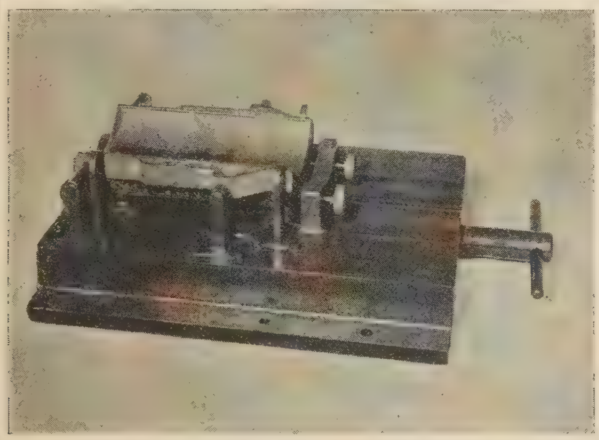

Fig. 13

between the service results of bearings unbroached, bearings improperly broached, and bearings properly broached. Obviously, until such performance data are available, conclusions as to the merits of broaching and the performance of the car journal bearing compared with other bearings have been founded on false assumptions.

In recognition of these facts, a prolonged study of the influence of broaching on the performance of car journal bearings was the initial work undertaken. The results of the study may be illustrated by the performance of three bearings selected at random from a lot of one hundred new bearings.

Fig. 7 shows these three $5^1/_2 \times 10$-in. A.A.R. bearings as received from the manufacturer unbroached, and Fig. 8, the same bearings operating under a total load of 16,375 lb; Bearing 1, was operated for 19 min at 30 mph, bearing 2 for 26 min at 30 mph, and bearing 3 for 30 min at 30 mph and for 9 min at 20 mph. These bearings were removed because of excessive heat and wiping of the journal lining. Temperatures of the bearings at the end of the periods of service were approximately 380 F. During the period of operation the bearings showed a frictional resistance of from 7.94 to 9.74 lb per ton.

Fig. 9 shows the same three bearings after the first test, broached in the usual manner. The bearings were again subjected to the same conditions of load at the same speeds. Bearing 1 failed after running 30 min at 30 mph and 10 min at 20 mph. Bearing 2 failed after running 30 min at 30 mph and 5 min at 20 mph. Bearing 3 failed after running 30 min at 30 mph and 3 min at 20 mph. The maximum period of service obtained from these bearings was 40 min. At the time of failure the bearings had reached a temperature of approximately 380 F. The effect of

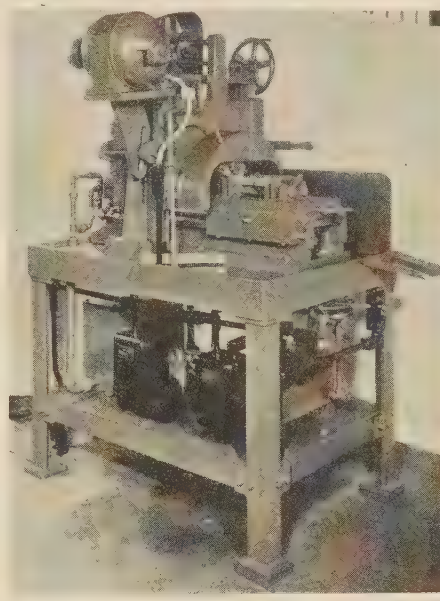

Fig. 14

the standard method of broaching resulted in frictional resistance during the interval of operation from 5.21 to 10.8 lb per ton. The appearance of the bearings on removal is illustrated in Fig. 10.

The same three bearings were again broached by a special method and their appearance is illustrated in Fig. 14. Again under the same conditions of load and speed these bearings were put in operation. Bearings 1, 2, and 3 were run for 30 min at 30 mph, 30 min at 20 mph, and 120 min at 50 mph. At the end of three hours the maximum temperature of these bearings was approximately 200 to 220 F. The frictional resistances recorded for this period ranged from 0.87 to 2.73 lb per ton. The appearance of bearings 1, 2, and 3 on removal is shown in Fig. 12.

The great improvement in the performance of bearing 3 in the last test, which was duplicated by bearings 1 and 2, might suggest that considerable expense would be involved to produce such results on all bearings. Fortunately, the final broaching was performed at no greater expense than is incurred in the standard method of broaching journal bearings.

Method of Broaching Bearings

The method by which the bearings were broached for the last test was adequate and proper. This method of broaching consists in placing the bearing face-up in a special chucking arrangement shown in Fig. 13, which insures that the bearing surface will be broached parallel to the back. The bearing is also broached on its exact center. The conditions of the bearings after service, as shown by a comparison of Figs. 8, 10, and 12, indicates the effect of this method of broaching as compared with the others. The bearing, held in this chuck, is mounted in a

broaching machine, illustrated in Fig. 14. The carriage supporting the chuck and its contained load is moved by a hydraulic feed. By means of a rapidly revolving cutter, the surface of the bearing is broached to a smooth, parallel surface of the desired journal radius. The head of the machine containing the rotating cutter

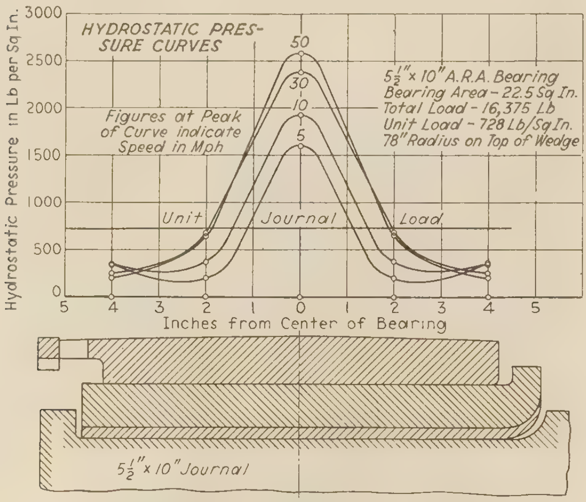

FIG. 15

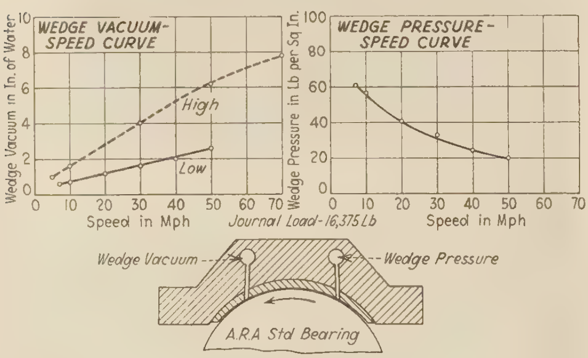

FIG. 16

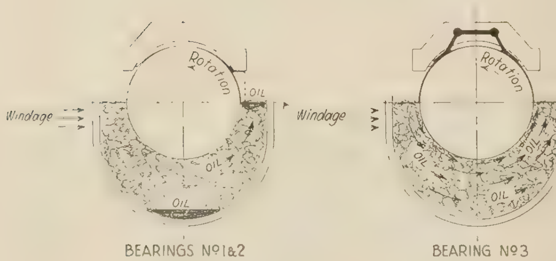

FIG. 17

head has motion in a vertical direction relative to the table so that the bearing may be broached to standard thickness. The final operation of cutting the fillet on the brass is performed automatically.

The conclusions drawn from the comparisons studied are that the proper broaching of bearings is essential to a low coefficient of friction and a satisfactory operation.

Load Conditions for Lubrication

With the variable of bearing surface eliminated, it is possible to investigate pressure and load conditions under which the bearing must be lubricated. The usual practice has been to divide the projected area of the bearing into the total load and consider the resulting load per square inch uniformly distributed over that area. By means of suitable drilling to pressure points and connections thereto of gages, the relation of pressure to speed under maximum load was determined. Fig. 15 illustrates a typical distribution of pressures as they vary longitudinally with speed and Fig. 16 the transverse distribution. While these tests were in progress, it was noted that the supply of lubricant to, and its return from, the bearing was as shown diagrammatically at

FIG. 18

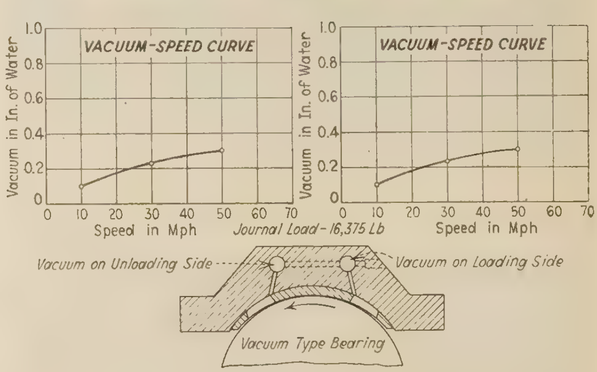

FIG. 19

the left of Fig. 17. Much speculation resulted from the data disclosed by these tests.

Fig. 18 shows the modifications of bearing construction that resulted from these speculations as well as certain modifications disclosed from an investigation of patent-office records and some bearings that were proposed by manufacturers.

Bearings 1 and 2 show the methods of obtaining film temperatures and bearings 3 to 7 the pressures. Bearings 8 to 16 show modifications of bearing area and oil circulation which finally developed bearings 18 and 19. Bearings 20 to 25 are suggested modifications. From this progressive process of investigation and elimination resulted the development of bearing 18.

Fig. 19 shows the characteristics of bearing 18.

The design of bearing 18 was perfected because of the following functional advantages over the present standard and other modifications tested. As will be seen from the right-hand figure of Fig. 17:

(1) There is sufficient oil stored in the bearing to operate for a protracted period without dependence on the packing.

(2) Lubricant is adequately distributed over the journal surface.

(3) Oil serves both as a lubricant and cooling medium for the bearing.

(4) The circulation of oil through the bearing promotes feeding from the waste.

(5) Foreign matter is prevented from getting under the bearing and from shutting off the oil supply at the rising side of the journal.

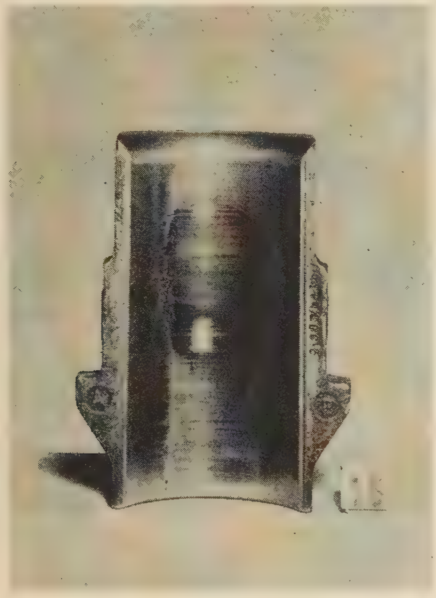

Fig. 20

(6) End leakage because of oil-wedge pressure is practically eliminated, for instead of pressure, as in Fig. 16, there is a vacuum, Fig. 19.

Over the last three years, several hundred bearings of various sizes in actual service in cars, locomotive tenders, engine and trailer trucks, and, in three cases, locomotive driving boxes oil lubricated, have substantiated the test-plant results.

Oils

The problem of determining the characteristics of the proper journal-box lubricating oil does not permit a complete discussion

Fig. 21

within either the scope of this paper or the present development of the art.

Generally it may be presumed that the tabulated specification requirements of the Association of American Railroads, operation and maintenance department, mechanical division, shown in Table 1 are representative of present practice. Four kinds of oil are covered, summer, winter, all-year, and reclaimed. The great majority of freight cars are lubricated with reclaimed oil. Oils 1, 2, 3, 4, 5, and 6 are oils that would meet the respective season specification with which they are grouped. Certain operating characteristics are shown for these six oils to illustrate the divergence of performance.

There may be a reasonable doubt as to whether the conventional specification constants are a measure of an oil as a lubricant, and whether they anticipate properly the requirements of service and differentiate adequately in insuring the degree to which the requirements will be met.

The necessity for two grades of oil, one for winter, the other for summer, or the substitution of a compromise year-around oil, is predicated entirely on the delayed feeding of oil from the waste at low temperatures. The

TABLE 1 COMPARISON OF SPECIFICATIONS FOR CAR JOURNAL LUBRICATION OIL

	Summer oil			Winter oil			All-Year oil			Reclaimed oil
	A.A.R.[a] spec.	Oil No. 1	Oil No. 2	A.A.R.[a] spec.	Oil No. 1	Oil No. 2	A.A.R.[a] spec.	Oil No. 1	Oil No. 2	A.A.R.[b] spec.
Flash (min), F	300	355	365	300	330	350	300	360	360	250 (min)
Saybolt viscosity										
At 210 F min desired	...	...	...	..	...	...	50	...	...	....
At 210 F min permitted	55	61	55	40	45	45	47	51	53	40 (min)
At 100 F max desired	450	...	...	260	...	...	...	...	...	....
At 100 F max permitted	725	500	289	300	160	142	300	290	279	725 (max)
Pour point test (max) F	+20	+10	—5	0	0	0	0	0	0	45 (max)
Water, per cent	0.10	Trace	Trace	0.10	Trace	Trace	0.10	Trace	Trace	0.5
Tarry matter, per cent	0.10	Trace	None	0.10	Trace	Trace	0.10	Trace	Trace	Not spec.
Insoluble impurities, per cent	0.10	Trace	Trace	0.10	Trace	Trace	0.10	Trace	Trace	0.5 (vol.)
Corrosion loss, 500-hr service										
Bronze, mg per sq dm	...	17.33	None	...	None	None	...	9.15	3.56	....
Babbitt, mg per sq dm	...	17.33	2.16	...	None	3.10	...	5.57	6.35	....
Steel, mg per sq dm	...	1.09	None	...	2.64	None	...	None	None	....
Tar formed, 500-hr service, by weight, per cent	...	0.68	None	...	Trace	0.02	...	1.35	Trace	....
Friction, kinetic, comparative										
At 5 mph, lb per ton	...	1.79	1.73	...	1.70	1.57	...	1.42	1.51	....
At 10 mph, lb per ton	...	1.70	1.23	...	1.52	1.22	...	1.38	1.44	....
At 30 mph, lb per ton	...	1.37	1.08	...	1.26	1.05	...	1.19	1.22	....
At 50 mph, lb per ton	...	1.37	0.94	...	1.16	1.10	...	1.15	1.06	....
At 70 mph, lb per ton	...	1.24	0.93	...	1.11	1.01	...	1.12	1.03	....

[a] A.A.R. spec. M-906-34.
[b] A.A.R. spec. M-904-30.

remedy for this lies more in the functional design of the bearing than in the characteristics of the oil or waste.

From a railroad operating standpoint seasonal changes of oil on all equipment to get full seasonal benefits are not physically possible. In reality boxes packed with winter oil operate in the summer and boxes packed with summer oil operate in winter. If the theory supporting seasonal oil is correct, most of the railroad journals operate most of the time with the wrong oil.

To illustrate the degree to which a change in bearing construction may insure successful lubrication, independent of the characteristics of the oil at low temperatures, tests were made.

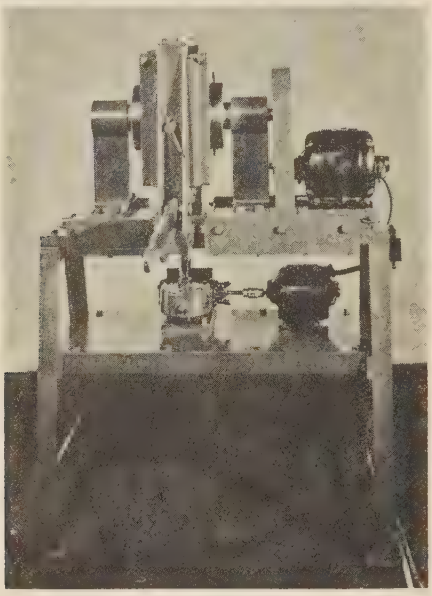

Fig. 22

Four oils of the specifications listed in Table 2 were used in packing the box of Fig. 9 with three kinds of waste, making twelve separate lots of packing.

TABLE 2

	Oil No. 1	Oil No. 2	Oil No. 3	Oil No. 4
Viscosity at 210 F......	54	46	54	51
Viscosity at 100 F......	650	215	383	331
Pour test, F............	+5	+0	+15	+10
Flash, F................	295	370	395	380

Twelve tests were made with bearing 18, Fig. 18, and twelve with bearing 1.

In each test the journal was operated at normal atmospheric temperature for a period; rotation was stopped under full load and the refrigeration unit turned on; temperature within the box reduced to —10 F and held more than 12 hr; after which, with the temperature at —10 F, operation was started at 30 mph.

Bearing 1 suffered a complete or partial failure in six out of the twelve tests. Those that were successful were with oil No. 3. A typical condition of failure is that shown in Fig. 20. Bearing 1 required some rebroaching after each of the twelve tests.

Bearing 18 operated successfully and normally throughout all twelve tests, and was not removed until all tests were completed. Fig. 21 shows the condition of the bearing on removal.

Waste and Packing

The use of inferior journal-box packing, oil-saturated waste, is a contributing factor of considerable consequence to failures in car journal lubrication, because reclaimed and renovated oil and waste are used to lubricate a large percentage of railroad rolling stock, particularly freight cars. The use of this material is justified only as an economy measure when the material is of a quality adequate for the demands of the service.

That there is not a general appreciation of the requirements of service is indicated by the existing A.A.R. specification for reclaimed oil, Table 1, and those regulations covering the quality of reclaimed packing. Under these existing specifications journal-box packing containing 3 per cent coarse dirt, 7.74 per cent fine dirt, 3.3 per cent moisture, 2.86 per cent tar, and 4 per cent metallic soap would meet the requirements and, therefore, be considered the equivalent of new waste and oil since the use thereof is permitted in lieu of new material. In a journal box containing 10 lb of journal-box packing there would be more than 2 lb of material of a nonlubricating nature and of a nature which would impede the feeding of the oil to the journal by the waste. This condition is possible because of the omission from the specifications of requirements which would limit or exclude the presence of such material from properly renovated or reclaimed journal-box packing. A specification should be written recognizing these elements of contamination and limiting them in amount to a maximum of one-fifth the respective amounts shown. Because of the large interchange of equipment between railroads the absence of an adequate specification to insure the use of proper journal-box packing by one road is reflected in the operating results of another.

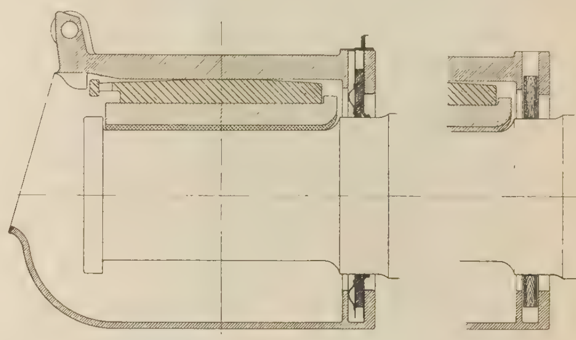

Fig. 23

In the service obtained from journal-box packing made of new waste and new oil the principal difficulties experienced with new waste are attributable entirely to improper methods of saturating the waste with the oil and the presence of lint in the new material, the removal of which greatly improves the functioning of the waste as a feeder of oil to the journal.

Dust Guards

Proper sealing of the journal box at the point where the journal enters the box is of importance, not only to keep the oil in the journal box but to exclude dirt and water as an element of contamination.

Investigation of the efficiency of various dust guards was carried on on the specially designed machine, Fig. 22. This machine consists of a rotating shaft extending through a box consisting of two dust-guard cavities so that two dust guards may be tested at one time. The shaft is the size of the axle entering a 6 × 11-in. journal box. The special box is filled with waste and oil at the proper saturation. The shaft is operated at a constant speed of 50 mph and at the same time moves back and forth through the box by an amount equal to the full permissible lateral motion of an axle. At the same time the box may be given

a vertical motion of varying amount. With this testing unit investigations were made of existing dust-guard constructions, of which style *A*, Fig. 23, is representative.

From the development work on this machine the sealing means or dust guard, style *B*, was developed. It was found with prevailing types of dust guards there was an inadequate seal, not only against the leakage of oil around the axle itself, but likewise to an even greater degree through and out of the dust-guard cavity at the bottom. Style *B* was developed in order to produce a gasket seal of a flexible and yielding structure against the walls of the dust-guard cavity and also an adequate seal around the axle fit. Fig. 24 illustrates the installation of the dust-guard seal in the present type of journal box before it is slipped over the axle.

Conclusions

The measure of the foregoing developments lies (1) in the degree to which continuity of operation is improved and (2) in the cumulative effect upon reduction in journal friction with the consequent resulting reduction in wear.

Fig. 24

As a measure of the accomplishment of the second objective, comparison is made of the journal running frictional resistance in pounds per ton of bearing 18 with the roller bearing under the

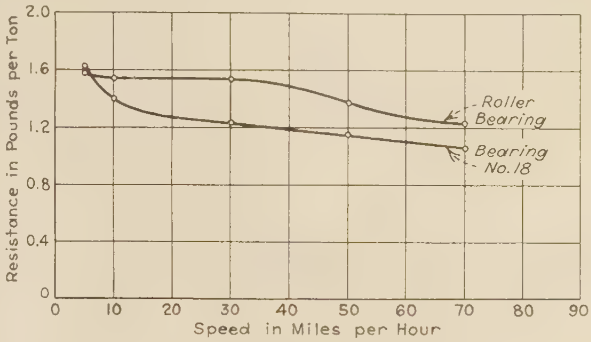

Fig. 25

same conditions of load, using the same oil (oil No. 2, Table 1), at speeds up to 70 mph, Fig. 25.

Continuity of operation by the elimination of lubrication failures lies in the proper preparation of packing, modification of bearing structure, and protection of these two by an adequate dust-guard seal.

Elimination of bearing failures, due to cracked, broken, or loose linings, cracked or broken bearings, lugs or collars, by the selection of the proper bearing back, lining metal, and the application of the lining to the bearing back, have been the subject of careful analysis. From this analysis has resulted the adoption from automotive practice of the forging process for the brass back instead of casting, and the centrifugal lining of the bearing instead of mandrel lining, and finally the selection of an improved babbitt metal.

By adoption of these developments, which retain all the structural elements of the existing equipment, journal operation will cease to be a limiting factor in the operation of railroad rolling stock and the direct and indirect expense now incurred will be greatly reduced.

A Study of the Turning of Steel Employing a New-Type Three-Component Dynamometer

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This paper is intended to present the results obtained in measuring with a new type of dynamometer the three components of the cutting force when turning with single-point tools. This dynamometer measures the cutting force as a function of the elasticity of its steel members. The forces are read on dial indicators and are not recorded graphically. The dynamometer is very rigid and simple in construction. It was developed by the authors to overcome many difficulties of calibration inherent in hydraulic recording dynamometers previously used by them.

In the turning of steel, the cutting force is conveniently resolved into tangential, radial, and longitudinal components. The tangential component accounts for practically all of the power required to remove the metal, the longitudinal component accounts for the power for feeding the tool along the cylindrical work, and the radial component holds the tool to the correct depth of cut and in so doing produces no power. The magnitude of these forces and their relation to each other varies widely for different metals. For a given material they change as the feed and depth of cut are changed, or as the tool shape is changed. Only slight changes result from a change in cutting speed.

This paper presents the results of a fundamental study of several phases of machinability in which the three components of the cutting force are determined for various tool shapes and sizes of cut. The built-up edge, its shape, size, and stability are studied as functions of the cutting speed. The temperature developed at the cutting point as affected by the cutting speed was determined by the tool-work thermocouple.

AN ACCURATE knowledge of the tangential, longitudinal, and radial components of the cutting force in the turning of steel and the variables affecting them is useful in designing machine tools, in selecting material for a part, or in the design of the part. Lighter, faster, more rigid, and more accurate machine tools are possible as a result of the use of these data in computing sections, etc. When a variety of materials will answer the purpose, a selection on the basis of cutting forces may increase production and accuracy. The part may be designed to withstand these forces, thereby reducing deflections caused by the forces on the tool.

Material and Equipment Used in Making the Tests.

The Material Used. A low-carbon steel twice annealed was used in the study discussed in this paper. A chemical analysis of the steel showed 0.21 per cent carbon, 0.27 per cent manganese, 0.030 per cent sulphur, and 0.022 per cent phosphorus. It was obtained as 6-in. square blooms in 4-ft lengths. Physical tests of the steel indicated a yield point of 19,900 lb per sq in., an ultimate strength of 54,000 lb per sq in., a reduction of area of 60 per cent, an elongation of 35 per cent in 2 inches, and a Brinell hardness of 99.

The Equipment. A 14-in. swing 16-speed, geared-head lathe 22 in., between centers, was used. It was equipped with a variable-speed, direct-current motor so that any possible cutting speed within the range could be obtained. A three-component-force dynamometer, designed and built by the authors, was mounted on the lathe in place of the compound rest. The machine with the dynamometer in place is illustrated in Figs. 1 and 2.

The construction of the dynamometer is simple. The axis of the central one-piece spool carrying the tool holder on one end is supported by two disks which are clamped securely at the periphery in the square housing. One disk is at the rear end of the axis, as seen in Fig. 2, and the second is at the forward end just back of the tool holder. The force on the tool bit supported in the holder at the forward end of the axis causes the axis of the spool to bend. The tangential and longitudinal components of the force are recorded on two dial gages mounted on the rear of the housing through a system of amplifying levers resting against the center of the axis of the spool. The radial force deflects the shaft along its axis by dishing the disks. A series of levers again transmits the deflection to the dial gage at the right. The whole device is locked into one rigid unit.

The dynamometer is simple, rugged, and foolproof. The calibration curves Fig. 3, show the relation between the force and gage readings. These are straight lines so the cutting force can be found by multiplying each dial reading by a given factor. These figures are 18.5 for the tangential, 27 for the radial, and 14.5 for the longitudinal forces. The dynamometer was calibrated up to 3200-lb tangential, 1800-lb radial, and 900-lb longitudinal force. It is probable that the dynamometer can be used for forces much higher than these, however, because of its rigidity.

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[2] Instructor in metal processing at the University of Michigan. Assoc-Mem. A.S.M.E. Mr. Kraus was graduated from the University of Michigan, College of Engineering, in 1932, and received his Master of Science degree in engineering in 1935. For a number of years he has worked with Professor Boston on various phases of metal cutting and is coauthor on several metal-cutting papers.

Contributed jointly by the Special Research Committee on Cutting of Metals and Machine Shop Practice Division and presented at the Annual Meeting of The American Society of Mechanical Engineers held at New York, N. Y., December 2 to 6, 1935.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1936, for publication at a later date.

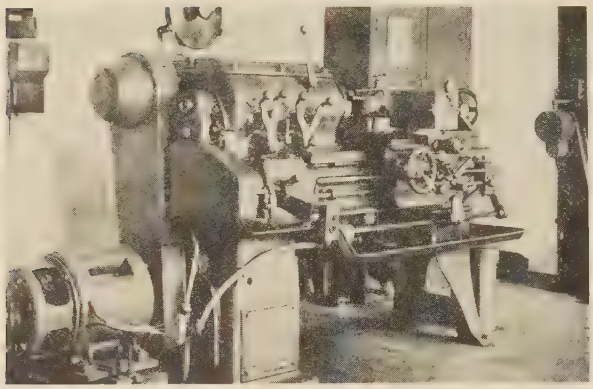

Fig. 1 Lathe and Three-Component Dynamometer Used in the Tests

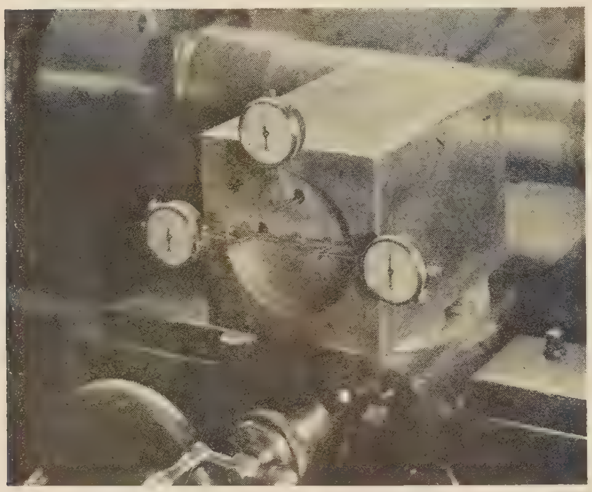

Fig. 2 The Three-Component Dynamometer

The maximum deflection of the cutting edge of the tool is about 0.002 in. The calibration curves apparently are unchanged by use, heat, or cutting fluid. Because of its weight and rigidity, chatter is practically eliminated. Several tool shapes which were impossible to test in a solid tool post of the lathe because of excessive chatter, operated without chatter when supported in the dynamometer.

The Tool Bits. High-speed-steel tool bits $^3/_8$-in. square of the 18-4-1 type were used. The tool shape selected as standard had an 8-deg back-rake angle; a 14-deg side-rake angle; a 6-deg front-and-side clearance angle; a 6-deg end-cutting-edge angle; a 0-deg side-cutting-edge angle giving a setting angle of 90 deg, since the axis of the bit was set at 90 deg to the axis of the work throughout the tests; and $^3/_{64}$-in. nose radius. The tool signature of this bit then is 8-14-6-6-6-0-3/64*R*, listing the angles in the order given. For simplicity, all tool shapes are referred to by their signatures. The sets of tool shapes tested can be classified as (1) those having variable side-rake angles of 0, 6, 14, and 22 deg (2) those having variable back-rake angles of 0, 8, and 14 deg, (3) those having variable side-cutting-edge angles of 0, 30, and 45 deg, and (4) those having variable nose radii of $^1/_{32}$, $^3/_{64}$, $^3/_{16}$, and $^1/_4$ in. The selected standard bit was common to all sets, making in all eleven tool shapes.

The bits were ground carefully on a special rebuilt cutter grinder using the face of a cup wheel. The surface produced was similar to a high-class surface grinder-finish, and the angles were accurate and reproducible to a fraction of a degree. All tests were run with the bits in the freshly ground condition, the bits being reground at the first sign of dulling. The tools were ground on the machine to a sharp point. The point then was ground to a radius by hand to conform to a standard radius gage. The radius and face then were honed lightly to remove burrs.

The Measurement of Cutting Forces

Cutting forces were recorded using the three-component dynamometer so that the tangential, longitudinal, and radial components could be determined separately for various values of feed and depth of cut for each of several shapes of tools. Values of feed and depth of cut were selected so the results could be plotted at approximately equal increments along the abscissa axis of log-log paper and at the same time cover a fairly wide range in values. The depths used in the tests were 0.030, 0.050, 0.100, and 0.150 in. Each depth of cut was run at four different feeds. The feeds selected were 0.0077, 0.0127, 0.020, and 0.030

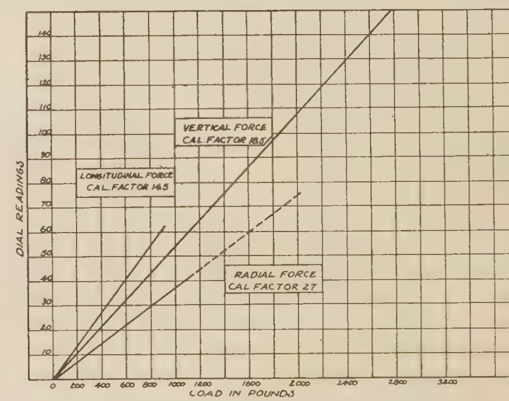

Fig. 3 Dynamometer Calibration Curves

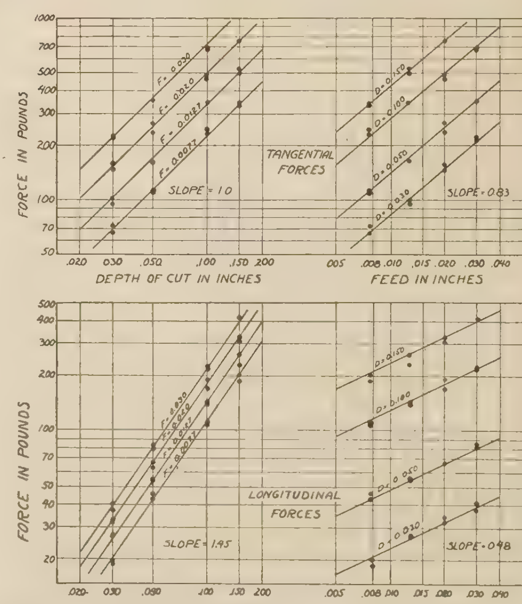

Fig. 4 Vertical and Longitudinal Force Test Data Plotted on Log-Log Paper to Determine the Exponents of the Cutting-Force Equation

(Annealed 0.21-C steel; cutting dry; tool shape 8-14-6-6-6-0-3/64*R*.)

in., respectively. This made 16-sizes of cuts taken by each tool shape.

The tangential and longitudinal forces obtained from the sixteen sizes of cut with the standard tool shape are shown plotted as ordinates on the log-log paper in Fig. 4. Values for constant feed over the variable depth are plotted at the left and values for constant depth over the variable feed at the right. Considering the tangential force, the highest line on the left represents the tangential force for all depths of cut except for a constant feed of 0.030 in.

TABLE 1 SUMMARY OF CUTTING-FORCE EQUATIONS

Tool no.	Tool signature	Tangential force	Longitudinal force	Radial force
1	8-0-6-6-6-0-3/64R	106500 $f^{0.74}d$	112500 $f^{0.54}d^{1.70}$	4670 $f^{0.76}d^{0.42}$
2	8-6-6-6-6-0-3/64R	137000 $f^{0.815}d$	61200 $f^{0.51}d^{1.55}$	1590 $f^{0.70}$
3	8-14-6-6-6-0-3/64R	133000 $f^{0.833}d$	33700 $f^{0.48}d^{1.45}$	923 $f^{0.56}$
4	8-22-6-6-6-0-3/64R	102500 $f^{0.80}d$	12600 $f^{.42}d^{1.35}$	704 $f^{0.46}d^{0.13}$
5	0-14-6-6-6-0-3/64R	156000 $f^{0.88}d$	51000 $f^{0.52}d^{1.58}$	2020 $f^{0.69}$
6	16-14-6-6-6-0-3/64R	94500 $f^{0.74}d$	28000 $f^{0.47}d^{1.38}$	416 $f^{0.47}$
7	8-14-6-6-6-30-3/64R	120000 $f^{0.80}d$	36500 $f^{0.60}d^{1.28}$	14500 $f^{0.48}d^{0.84}$
8	8-14-6-6-6-45-3/64R	88000 $f^{0.74}d^{0.93}$	31400 $f^{0.66}d^{1.12}$	40700 $f^{0.77}d^{1.00}$
9	8-14-6-6-6-0-1/32R	137000 $f^{0.84}d$	34300 $f^{0.48}d^{1.45}$	692 $f^{0.53}$
10	8-14-6-6-6-0-3/16R	58000 $f^{0.68}d^{0.83}$	31800 $f^{0.57}d^{1.31}$	7250 $f^{0.68}d^{0.47}$
11	8-14-6-6-6-0-1/4R	41000 $f^{0.59}d^{0.79}$	30000 $f^{0.63}d^{1.22}$	14600 $f^{0.84}d^{0.43}$

NOTE: The material cut was annealed low-carbon steel, when using 3/8-in. square high-speed-steel tool bits. The cutting was done dry at approximately 80 fpm on a bar from 4 to 6 in. in diameter.

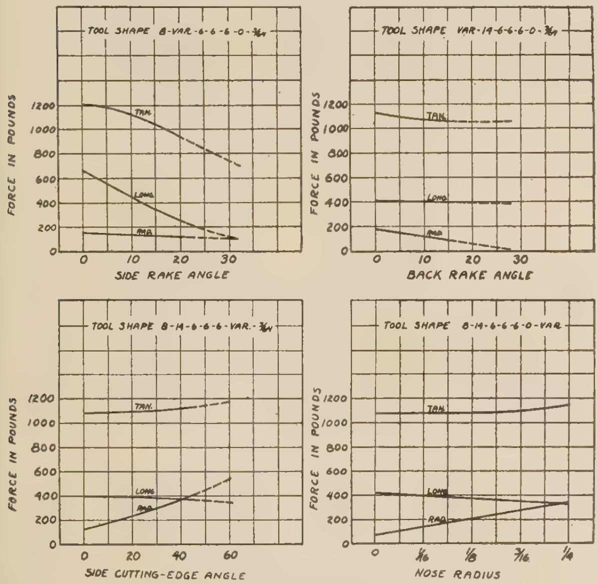

FIG. 5 EFFECT OF TOOL SHAPE ON THE THREE COMPONENTS OF CUTTING FORCE

(Annealed 0.21-C steel; cutting dry; feed 0.030 in.; depth 0.150 in.)

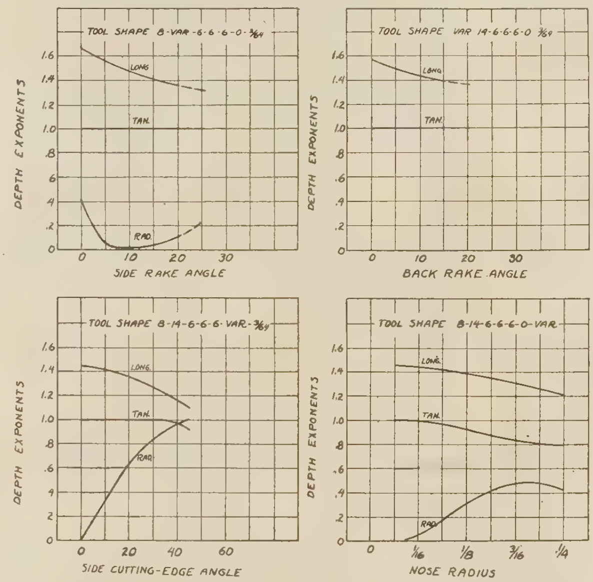

FIG. 7 EFFECT OF TOOL SHAPE ON VALUES OF DEPTH EXPONENTS IN CUTTING-FORCE EQUATION $F = Cf^ad^b$

(Annealed 0.21-C steel; cutting dry.)

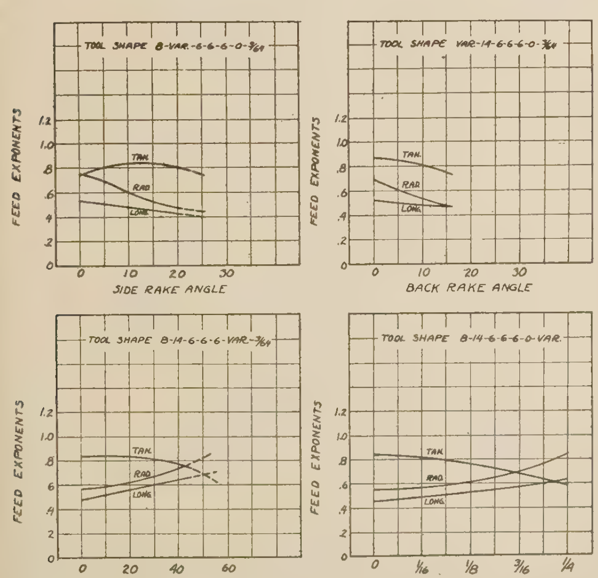

FIG. 6 EFFECT OF TOOL SHAPE ON VALUES OF FEED EXPONENTS IN CUTTING-FORCE EQUATION $F = Cf^ad^b$

(Annealed 0.21-C steel; cutting dry.)

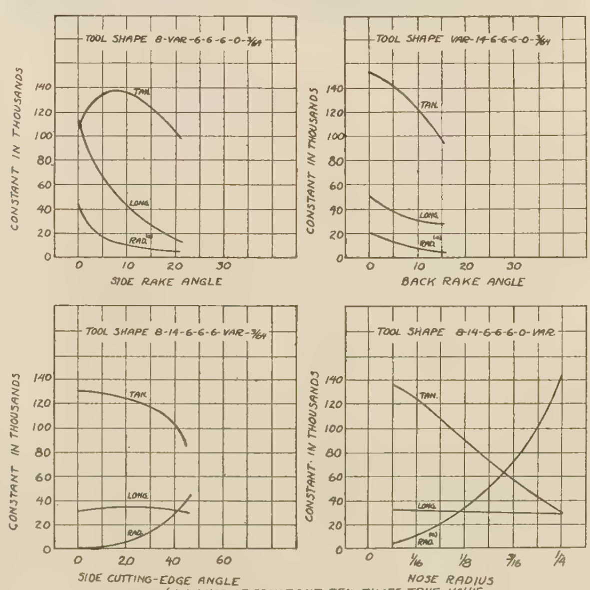

FIG. 8 EFFECT OF TOOL SHAPE ON VALUES OF CONSTANTS IN CUTTING-FORCE EQUATION $F = Cf^ad^b$

(Annealed 0.21-C steel; cutting dry.)

The highest line of the tangential force on the right represents those forces obtained for various feeds at a constant depth of cut of 0.150 in.

It is convenient that these data may be represented by straight lines. The force equation, as a function of variable feed and variable depth, considering again the tangential force only as shown as the highest lines in Fig. 4, may be expressed as

$$T = Cf^a d^b$$

where T is the tangential component of the cutting force; C is the constant depending upon the tool shape, material cut, cutting fluid, etc.; f is the feed in inches per revolution; d is the depth of cut in inches; and a and b are the exponents to be determined.

Since f and d are independent variables, one must be held constant while the other is varied in order to determine its effect. Holding the feed f constant in studying the influence of variable depth d would change the equation to

$$T = Kd^b$$

where K is a new constant equal to Cf^a. Plotting the tangential forces for a given feed at various depths, as shown by each of the four lines in the upper left of Fig. 4, the following equation is obtained

$$\log T = b \log d + \log K$$

This represents the equation of a straight line of the form

$$y = mx + c$$

where m corresponds to b and is the slope of the line. The slope also may be represented by the tangent of the angle made with the horizontal.

The feed exponent is obtained in a like manner. Also, corresponding equations for L and R, the longitudinal and radial components, respectively, are similarly determined.

The equations for the three components of cutting force for each of eleven tool shapes are summarized in Table 1. In order to visualize the effect of the various tool-shape angles, Figs. 5, 6, 7, and 8 have been prepared. Fig. 5 shows actual cutting forces for a feed of 0.030 in. per revolution and a depth of cut of 0.150 in., plotted over various values of the variable applied to the standard tool.

The curves shown in Fig. 5 would have different forms for any other size of cut, inasmuch as the feed and depth exponents are not the same for the different tool shapes. It will be noticed that the tangential and longitudinal forces drop considerably with increasing side rake, as shown at the upper left in Fig. 5, but drop very little for increasing back rake, as shown at the upper right of the same figure. The radial force is affected but little by a change in side rake, but is greatly decreased by increasing the back rake. Increasing the side-cutting-edge angle or the nose radius seems to give the same effect on the forces. The tangential component is increased slightly, the longitudinal component is decreased slightly, and the radial component is increased markedly, as the side-cutting-edge angle and nose radius are increased, as shown in the lower part of Fig. 5.

Fig. 6 shows the values of the feed exponent plotted over the tool-shape variables applied to the standard tool. A rather surprising range of values is found in nearly all cases, showing that a change in tool shape materially influences the value of the feed exponent. By changing the exponent of the feed in the force equation, the values of the forces themselves are changed. Obviously, for this material the optimum tool shape from the cutting-force standpoint may be quite different for each of several different feeds.

Fig. 7 shows a similar set of curves indicating the wide range of values of depth exponent as a function of the tool-shape variables. Peculiarly, the depth exponent of the tangential component is affected but little by any changes until large side-cutting-edge angles or nose radii are used. The radial force is not influenced by depth for a number of tool shapes. The radial forces for variable depths of cut were, however, too erratic in many cases to obtain a very accurate exponent. Fig. 8 shows the effect of the tool-shape variables on the constants of the three equations. The values of the exponents of the feed and depth also materially influence the values of the constants.

Inasmuch as the 0.22 per cent carbon steel was soft, the force values represented by equations in Table 1 might not be applicable to the higher-strength steels. For this reason, formulas for the tangential, longitudinal, and radial forces are given in Table 2 for two commonly used tool shapes when cutting an annealed S.A.E. 3135 steel. The force values and equations for tool No. 1 of Table 2 may be compared with tools Nos. 3 and 7 of Table 1 used in cutting the low-carbon steel. For the 15-deg side-cutting-edge tool, when cutting the low-carbon steel, the values of force in pounds, feed exponent, depth exponent, and constant may be read directly from the curves in the lower left-hand corner of Figs. 5, 6, 7, and 8, respectively. For instance, when cutting the low-carbon steel, the force for the 15-deg side-cutting-edge tool is found to be about 1090 lb, from the lower left corner of Fig. 5, as compared with 1250 lb for tool No. 1 in Table 1. The respective longitudinal and radial values are 400 lb as compared with 480 lb, and 200 lb as compared with 320 lb. The S.A.E. 3135 steel probably represents an average difficult machining steel, whereas the low-carbon steel represents one of the easiest machining steels.

It is interesting to record Taylor's formula for cutting an 0.34 per cent carbon steel annealed to a tensile strength of 70,280 lb per sq in. The equation for tangential cutting force was $T = 230{,}000\ df^{0.934}$. For hard steel, the constant was 296,000. Taylor used his round-nosed forged tools with 8-deg back rake and 14-deg side rake, when cutting at 60 fpm.

TABLE 2 CUTTING-FORCE AND FORCE EQUATIONS[a]

	Tool No. 1	Tool No. 2
Tool signature	8-14-6-6-6-15-3/64R	8-22-6-6-6-15-3/64R
Tangential force	259000 $f^{0.98}d^{1.0}$ 1250 lb	174000 $f^{0.90}d^{1.0}$ 1110 lb
Longitudinal force	35700 $f^{0.39}d^{1.55}$ 480 lb	950 $d^{1.06}$ 123 lb
Radial force	7300 $f^{0.74}d^{0.28}$ 320 lb	25700 $f^{0.84}d^{0.58}$ 450 lb

[a] When turning an annealed S.A.E. 3135 steel dry at 50 fpm with 3/8-in. square high-speed-steel tool bits. Force values listed are for a depth of cut of 0.150 in. and a feed of 0.030 in.

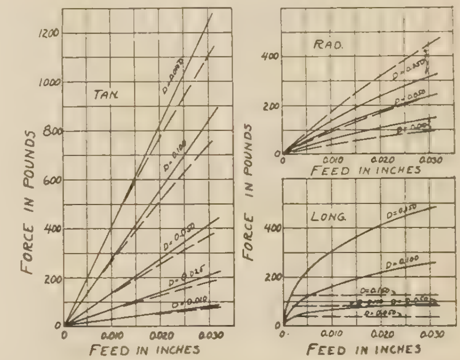

Fig. 9 Tangential, Longitudinal, and Radial Forces for Various Depths of Cut and Feeds When Machining Annealed S.A.E. 3135 Steel With a Dry Cut at 50 Fpm and With 3/8-In. Square High-Speed-Steel Tool Bits

(The solid lines are for tool No. 1 of Table 2 which has a 14-deg side-rake angle and a 15-deg side-cutting-edge angle. The dashed lines are for tool No. 2 which has a 22-deg side-rake angle and a 15-deg side-cutting-edge angle.)

Fig. 9 has been prepared to show better the values of the cutting force for both tools in Table 2. This gives the cutting forces for various combinations of depth of cut and feed. On the left is shown the tangential cutting forces for any value of feed up to 0.030 in. for five different values of depth of cut. The solid line represents the cutting force for tool No. 1 having the 14-deg side-rake angle, and the dashed line represents the cutting force for the corresponding conditions for tool No. 2 having a 22-deg side-rake angle. In the lower left are shown the values of the longitudinal component of the cutting forces for any feed at the several depths. Again the solid lines represent the cutting force for tool No. 1, whereas the dashed lines represent constant cutting values for longitudinal force which are obtained with tool No. 2 having 22-deg side rake. The longitudinal-force equation for this tool as given in Table 2 shows that the exponent of f is zero. In the upper right of Fig. 9 are shown the values of the radial component for both tools for each of several depths for all feeds up to 0.030 in. These curves show that the radial force for tool No. 2, having 22-deg side rake, is low for low values of depth, but is comparatively high for the higher values of depth.

Influence of Cutting Speed on Cutting Forces

All of the preceding tests were run at a cutting speed of 70 to 80 fpm. Because of the soft free-cutting nature of the steel, a considerable range of speed was possible. Therefore, to determine the cutting-force equation, a series of tests was run with one tool shape at surface-cutting speeds of 26, 31, 42, 56, 70, 90, 115, 150, 210, 240, and 320 fpm. The signature of the tool bit selected was 8-14-6-6-6-30-3/64R, the 30-deg side-cutting-edge angle

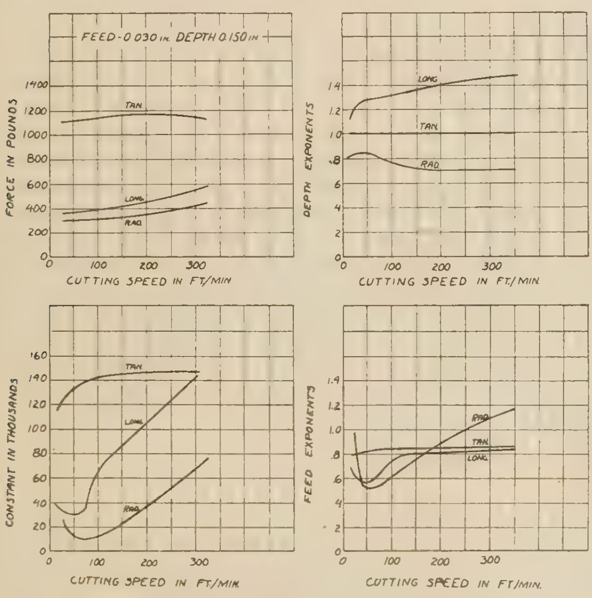

Fig. 10 Effect of Cutting Speed on Forces, Constant, and Exponents

(Annealed 0.21-C steel; cutting dry; tool shape 8-14-6-6-6-30-3/64R.)

being selected instead of the standard tool because of the higher allowable cutting speeds for the same tool life.

At speeds below 50 fpm, the results were unexplainably erratic until it was noticed that quite a large built-up edge was adhering tightly to the tool bit as the cuts were changed from one size to another. Different force values were obtained if the order of the cuts was changed. This indicated that the built-up edge did not quickly change its size and shape as the cutting conditions were changed. The bar was soft and the built-up edge was hard due to

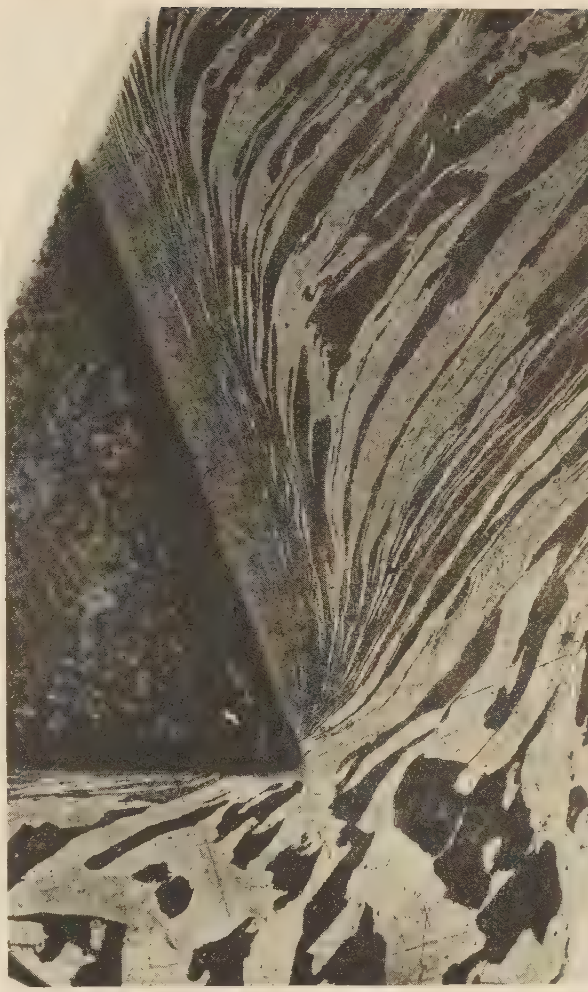

Fig. 11 Photomicrograph of Cross Section of Built-Up Edge Showing Flow of a Chip of Low-Carbon Steel Annealed to Give a Brinell Hardness of 99

(Tool signature, 8-14-6-6-6-30-3/64R; cut taken at 0.03-in. feed and 0.15-in. depth. The cutting speed was 200 fpm. From V. Prianishnikoff.)

cold working. Knocking off these built-up edges too often chipped the keen cutting edge of the tool. The most practical method of reducing this effect was found in changing bits for each depth, and then running the feeds in order of increasing size. This did not eliminate the effect entirely, however, and the data for the slower speeds are not so consistent as for the faster speeds.

The results of this series of tests are shown graphically in Fig. 10. The cutting forces are plotted against cutting speed for the 0.030-in. feed per revolution and 0.150-in. depth of cut. The peculiar shapes of the exponent and constant curves for the slower cutting speeds are undoubtedly due to the built-up edge, but the cutting-force curves show no effect at all. As would be expected, cutting speed has very little effect on tangential forces, exponents, or constants. Disregarding the slower speeds, it will be noticed that the depth exponent in the longitudinal-force equation increases somewhat with an increase in speed. The exponent for the feed remains nearly constant, but the values of the constant increase very rapidly as the speed increases. In the radial-force equation, the exponent of depth remains nearly constant, the feed exponent shows a marked increase in value as the speed is increased, and the constant increases rapidly as the speed is increased.

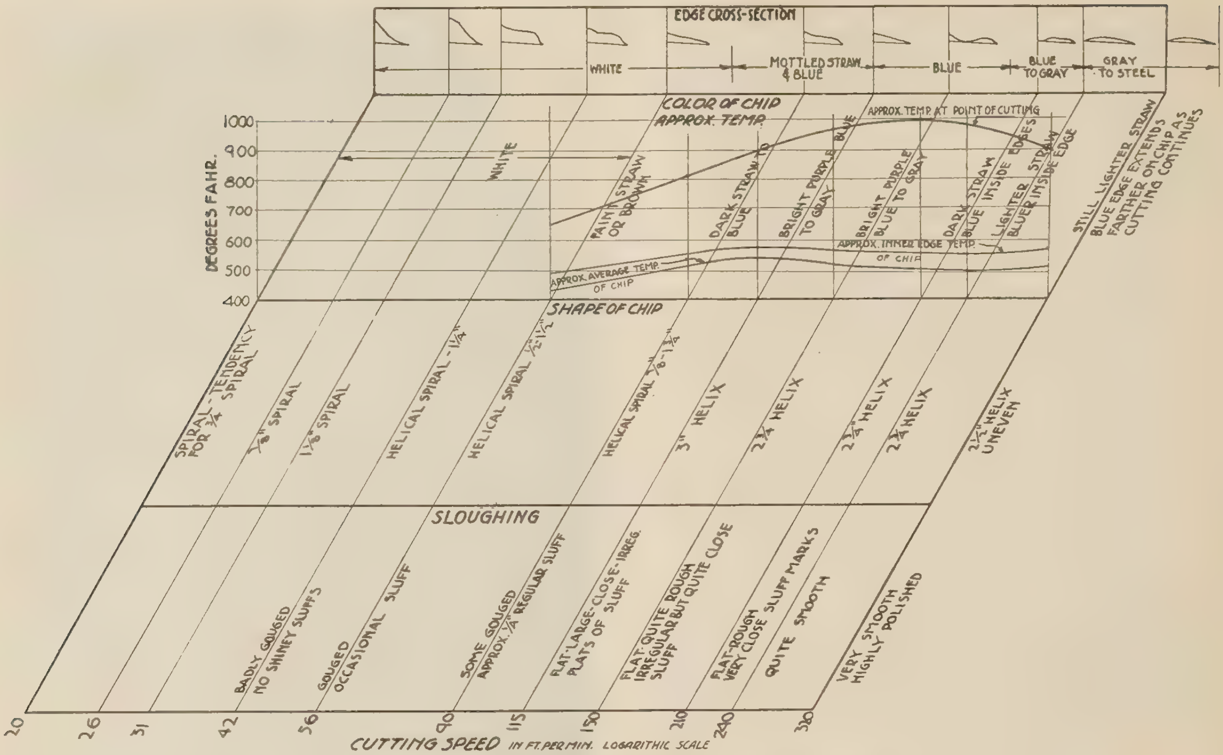

Fig. 12 Observations of Built-Up Edge at Different Speeds

The Built-Up Edge

The characteristics of the built-up edge formed on the tools during these tests proved to be interesting. The edge is often found adhering to the steel chip at the end of a planer, shaper, and miller cut. It is, however, not so obvious in turning operations where the cuts are continuous. The built-up edge formed by this soft ductile steel was much harder than the original bar and was unusually large for steel. This permitted an exceptional opportunity to observe its size and shape under various cutting conditions. Admittedly, the best method of obtaining a true picture of the built-up edge is to stop the work suddenly in the cut. This has been done in another series of tests in which the metal and head of the chip containing the built-up edge were cut from the bar and sectioned, polished, and photographed at various places. Fig. 11, produced by this process, indicates the flow of the chip and the character of the built-up edge, and represents the condition of the edge obtained at about 200 fpm cutting speed with this soft steel for the standard tool shape.

It was discovered in cutting this soft steel that when the tool bit was quickly withdrawn from the cut, the built-up edge remained apparently undisturbed on the tool face. The observations of the built-up edge at different speeds are recorded in Fig. 12. Arranged on a logarithmic scale at the top of the figure is shown a diagrammatic section through the center of the built-up edge when the tool was quickly withdrawn from a cut of 0.030-in. feed and 0.150-in. depth at various cutting speeds. The tool shape is the same as that indicated in Fig. 11. The change in the shape and size of the built-up edge throughout this speed range is very great. It is thick and sharp at the low speeds and thin and flat at the high speeds. At the higher speeds, particularly above 200 fpm, the edge seems to be formed on the face of the tool back of the cutting edge, leaving a small rim of the face next to the cutting edge exposed. This condition is shown in Fig. 11. The drawings of Fig. 12 correspond to the dense portion of the chip of Fig. 11.

The shape of the chip also undergoes a change in shape as the speed is increased. The chips are spiral in shape up to about 30 fpm. They then become helical for the higher speeds. For speeds from about 42 to 90 fpm, the chips have a helical spiral form as indicated horizontally across the central part of Fig. 12. At the low speeds there appears to be no sloughing of the built-up edge. The underside of the chip is very rough. At about 56 fpm an occasional portion of the built-up edge slides or sloughs off as a part of the underside of the chip. These sloughs increase in number and become more irregular until at high speed the slough is continuous and the underside of the chip is very smooth and highly burnished as indicated in the lowest section of Fig. 12. At about 90 fpm the underside of the chips carried even and uniformly spaced sloughs.

Cutting Temperatures

While running the variable-speed tests using a feed of 0.030 in. and a depth of 0.150 in. with the standard shape of tool listed in Fig. 11, temperatures as obtained by the tool-work thermocouple were recorded. The work and tool were calibrated in an electric furnace so as to determine the electromotive force developed at their point of contact at different temperatures. The calibration curve had temperature values of 70, 200, 400, and 600 F for 0, —0.35, —0.45, and 0 millivolt, respectively, and approximately a straight line 1000 F giving plus 1.4 millivolts. All measurements taken were above 600 F so the negative loop in the curve was not used. The temperatures in degrees Fahrenheit are shown plotted over the cutting speeds arranged on logarithmic scale near the top in Fig. 12. At 56 fpm the cutting temperature was approximately 650 F. It corresponded to a maximum of 1000 F at 210 fpm, and fell off to about 900 F at 320 fpm. At this highest

speed the tool life was about 1 minute, showing that the actual temperature of the tool at its cutting edge may have been well above the 900 F recorded by the tool-work thermocouple. The most probable cause for the drop in the temperature curve at the highest speed is the influence of the built-up edge which, at the higher speeds, protects the tool face almost entirely from the flowing chip. At lower speeds the chip slides over the built-up edge and rubs over the tool face. In other tests on a 0.61-per cent carbon steel, the temperatures developed at the tool point, as measured by the tool-work thermocouple, were found to increase in direct proportion to the cutting speed between 550 F and 1100 F.

Plotted on the same coordinates in Fig. 12 just below the temperature curve are two curves giving the approximate temperature of the chip, as indicated by the temper color. The higher curve gives the temperature of the chip formed at the bottom of the cut. This temperature varied only about 5 per cent between cutting speeds of 100 and 320 fpm. Excess heat probably was carried away by the fast-moving chip.

Conclusions

A number of conclusions may be drawn from the foregoing study and data. While these conclusions hold only for the material cut and under the conditions of the test, it is believed that many of the trends revealed are true of the operations of single-point tools in general.

1 Equations of the form $T = Cf^a d^b$ are found to hold for all three components of the cutting force, namely tangential, longitudinal, and radial. The values of C, a, and b will vary materially for different shapes of tools as indicated in Table 1.

2 The tangential cutting force is reduced uniformly to about 40 per cent by increasing the side-rake angle from 0 to 30 deg.

3 A change in back rake, side-cutting-edge angle, and nose radius has little effect on tangential forces or longitudinal forces.

4 The longitudinal force almost disappears with high values of side-rake angle.

5 The radial force is little affected by side rake, drops to zero with high values of back rake, and increases rapidly with higher values of side-cutting-edge angle and larger nose radii.

6 The exponent of the feed in the tangential-force equation first increases and then decreases with increasing side-rake angles. The highest value of 0.82 is for 12-deg side rake.

7 An increase in back rake, side-cutting-edge angle, or nose radius tends to reduce considerably the feed exponent in the tangential-force equation.

8 Increasing either the back- or side-rake angle decreases the feed exponents for both the radial and longitudinal forces, while an increase in side-cutting-edge angle or size of nose radius increases these exponents.

9 The depth exponent of the tangential-force equation is unaffected except for large values of side-cutting-edge angle or large nose radii, when it is reduced somewhat.

10 An increase in back rake, side rake, side cutting-edge angle, or nose radius decreases the depth exponent of the longitudinal-force equation.

11 An increase in the side-cutting-edge angle or nose radius causes a very large increase in the depth exponent of the radial force.

12 Very pronounced changes in the values of the constant in all equations result from changes in any of the tool angles.

13 The tangential force remains practically unchanged when the cutting speed is increased from 26 to 320 fpm.

14 Both longitudinal and radial forces increase somewhat with an increase in cutting speed.

15 At speeds below 100 fpm, the exponents of feed and depth and the values of the constant were affected considerably by a change in speed. This was believed to be due to a change in the shape of the built-up edge.

16 The cross section of the built-up edge changes radically as the cutting speed increases. At low speeds it is large with a steep angle. At higher speeds it is flatter, and at speeds above 200 fpm, it tends to desert the cutting edge and occur as a flat thin patch.

17 At slow speeds, the chip comes off in the form of a spiral which gets larger and looser as the speed increases. This changes to a helix of about 3 in. in diameter at 115 fpm. At higher speeds, the helix decreases in diameter somewhat and becomes more uneven.

18 During the cutting action, more or less sloughing of the built-up edge occurs, the sloughs remaining on the underside of the chip. The sloughs first appear at about 50 fpm and then, as the cutting speed increases, they increase in number until at high speed the under surface is one continuous slough and is smooth and highly polished.

19 Using the tool-work thermocouple method, the temperature at the face of the tool was found to rise to 1000 F at 210 fpm and then drop to 900 F at 320 fpm for a given tool shape and size of cut.

20 The temperature of the chip as determined by temper colors was changed less than 5 per cent in the range of cutting speed from 90 to 320 fpm.

Discussion

Principles Underlying the Rational Solution of Automatic-Control Problems[1]

P. S. Dickey.[2] The writer finds it necessary to question many statements made in Mr. Mitereff's paper and believes that the terms used therein should be properly defined and classified before a rational solution is attempted.

Mr. Mitereff says that the system external to the regulator can be divided into four factors: (*a*) storage of fluid or energy, (*b*) inflow of fluid or energy to storage, (*c*) outflow of fluid or energy from storage, and (*d*) a function indicative of the amount of fluid or energy in storage.

Clearly, this factor (*d*) should be classified separately since it is only a metered indication of factor (*a*) and therefore should be placed on a parity with metered indications of factors (*b*) and (*c*), which are the inflow and outflow of fluid or energy from storage. One, or more than one, of the metering indications may be used in developing the control apparatus.

Mr. Mitereff divides his control apparatus into three parts, but although it is a reasonable division it is not in agreement with customary practice as described later. However, with all the mathematics which the author applies to the devices expressed as "operative connections between the impulse-receiving element and the final-operating element," he completely neglects the equation between motion of the final-operating element and the actual delivery of energy or fluid to or from storage. Every one who has had much practical experience in adjusting automatic-control systems knows that this equation is of utmost importance.

The writer would further like an example of a control system in which the final operating element is an electric contactor as this does not seem consistent with the final operating element as defined.

It is convenient to think of control systems as being divided into four parts: (1) The measuring device which corresponds to Mr. Mitereff's impulse-receiving element; (2) the operative connection which may consist of pilot valves, electric contactors, mechanical linkage, and provides the desired motion characteristic and/or the necessary power amplification between the measuring device and the power device described in (3); (3) the power device. This may consist of a piston, a reversing pilot motor, a diaphragm motor, or other device capable of providing the necessary work to actuate item (4); and (4) the regulatory device. This may consist of a valve, a damper, a rheostat, or any other device of this nature capable of regulating the flow of fluid or energy.

This division of the control system allows segregation of the equation of the control system from the equation of the regulatory device which often is not furnished by the control manufacturer.

Mr. Mitereff's statement regarding "the insignificance of sensitiveness and sticking of measuring and control apparatus" clearly indicates lack of experience. No control can be better than its measuring device. Instrument makers have spent years improving metering devices from which automatic controls are actuated. Friction and inertia of moving parts are the worst enemies of the control manufacturer and no amount of mathematics will eliminate or compensate for them.

Undoubtedly, time lag is a serious obstacle to successful adjustment of any control installation, though there remains considerable confusion regarding system time lag and system storage of energy or fluid. Mr. Mitereff's choice of a pressure regulator as an example of distortion of perfect correspondence of the actual pressure, and of the pressure at the regulator, is somewhat unfortunate since many regulators use Bourdon tubes wherein the displacement of fluid in the pilot line is negligible, so that the impulse travels along the pilot line approximately at the velocity of sound. In general, temperature regulators are much more likely to be troubled by the distortion described. Likewise, the time lag between the instant of increase of fuel and air supplied to a boiler furnace and the instant of increase in steam generation of the boiler is a much more effective example than that of the hydraulic turbine.

Referring to Mr. Mitereff's example of the direct-acting float-operated valve controlling discharge of fluid from a tank, the writer must call attention to the real problem involved, namely controlling fluid output in accordance with fluid input. The control which supposedly solves the equation for the tank level does not necessarily accomplish the ultimate aim. Furthermore, there are often variations of this level-control problem not mentioned by the author. The following examples are quite prevalent:

1 In oil-refinery work, the flow to bubble towers often is highly variable and a successful control must allow the level to fluctuate and thereby utilize the storage capacity of the towers to maintain an output which averages the input fluctuation.

2 In steam boilers the drum level is affected not only by the input of water and outflow of steam but by the volume occupied by steam bubbles below the water level, so that changes in water level are not necessarily an indication of the outflow–input ratio. The proposed equations for obtaining water level would be useless in either this or the preceding case.

Mr. Mitereff's paper would make easier reading if he were more consistent in his use of symbols. For instance, in one case F is the distance traveled by the valve, and immediately following F becomes the final regulating effect of the automatic-control apparatus. Does this final symbol mean a motion or a change in energy or fluid flow?

In Equations [1] to [6], if the author means F to represent the motion traversed by the regulating device, the writer agrees with these equations except that the regulator illustrated and identified as an example of Class-I regulators should not be called the most common type. Much more extensive use is made of the Class-II regulator, the characteristics of which the author gives in his Equation [2].

The writer believes that an important control method has been eliminated from the author's list of control types. It is the three-element control which measures both input and output of fluid or energy, and maintains them approximately equal with only a relative minor adjustment of either from a measure of the storage of the system.

The application of this control system to feedwater regulation

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[1] Published as paper FSP-57-9, by Sergei D. Mitereff, in the May, 1935, issue of the A.S.M.E. Transactions.

[2] Research Engineer, Bailey Meter Company, Cleveland, Ohio. Jun. A.S.M.E.

for high-capacity steam boilers is described by M. F. Behar[3] and in bulletins published by the Bailey Meter Company.

Other successful applications of this three-element control system have recently been made and it is the writer's prediction that extensive use of this system will be made in the future in control problems which cannot be solved by other methods.

M. J. Zucrow.[4] With the field of application for continuous automatic controllers becoming increasingly more extensive, it is essential that the manufacturers and users of such devices have a common nomenclature to avoid the confusion of ideas which inevitably arises when the terminology of a subject is used indiscriminately. Consequently, writers on this subject should adhere to an accepted nomenclature as far as possible, and to define accurately their terms when no accepted terminology exists. Mr. Mitereff's paper is a timely contribution to the literature on this subject, but his unorthodox use of terms is regrettable.

Continuous automatic-control mechanisms, such as fluid-flow controllers, are offshoots of that class of controllers known as speed governors. They embody the application of the fundamentals of speed control to other control problems. The literature pertaining to the theoretical developments of governing dates back to 1840.[5] The first theoretical solution which took into account the reactions between the governor and prime mover is due to Wischnegradsky (1877).[6] Since that time there has been a more rapid advance in the coordination of the theory and practice of this art. In more recent years the published works of such engineers as Tolle, Stodola, Wunsch, Stein, Hodgson, Smith, Mason, and others have done much to clarify the control problem. The dynamics of pilot or relay speed governing has been investigated thoroughly by Dr. Tolle.[7] By properly modifying his equations, in accordance with the method developed by Stodola,[8] the speed-governing equations of Tolle can be applied to the mathematical analysis of such complicated problems as those which arise in complete combustion control.[9] It is, therefore, surprising to find, with Dr. Tolle's work listed in Mr. Mitereff's bibliography, that the latter's paper contains such statements as (1) "little has been published relative to the fundamentals" of governing and control; (2) there is "an absence of data necessary for rational solutions;" and (3) "the concept of the characteristic of a regulator or governor . . . has received little attention."

The elements of a continuous automatic-control installation have been described by Mr. Mitereff. However, due to his uncommon use of terms, a brief repetition of these elements, using the more common terminology, will help in overcoming any confusion which might otherwise arise. Every automatic controller comprises a primary system, which may be regarded as a measuring instrument, and an operative means for effecting corrections to the function being controlled. The particular physical arrangement and function which it is to control, is called the application. The required flow of fluid or energy which the physical system needs in order to maintain the controlled function at its normal value is called the demand, and that furnished to the system is known as the supply. For the controlled function to experience no "departure from normality" the supply must equal the demand and the capacity or storage of the application must also be at its normal value. Consequently, for the application to be susceptible to automatic control, the function to be controlled must be measurable, the primary element must be installed adequately, and there must be sufficient power available for actuating the control device.

Automatic control devices may be conveniently divided into two main groups, which are (1) self-operating controllers which furnish direct control, and (2) pilot or relay controllers which furnish pilot or relay control.

A self-operating controller is one which operates a valve (or other device) directly from its primary element without the assistance of additional forces from an outside source of power; the primary element and the valve are connected together and the latter moves in direct proportion to the departures from normality of the primary element. A pilot controller is one in which the action of the primary element operates a pilot (or pilot valve) which releases a source of power, such as compressed air, to a power device (piston) for actuating the valve; the power piston may then be located at a distance from the rest of the control apparatus. The second class of controllers comprises by far the majority in industry. While it is not always necessary, pilot controllers are usually equipped with some device which ties together the movement of the power piston and the pilot valve. Such an element is called a "follow-up" or a "return motion." It usually consists of a linkage between the power piston and the pilot valve, whereby the latter is returned to its neutral position when the controlled valve has been moved through the correct distance.

The analysis of controller problems shows that time differences arise between the correspondence of cause and effect throughout the system. These time differences are of two kinds: (1) Those inherent to the automatic controller and (2) those inherent to the application. The former are termed the "controller period" and the latter the "application lag" or "time lag."

The controller period is the sum of the time differences pertaining to each element of the automatic controller. M. F. Behar[3] in discussing controller period, has pointed out that "in a pressure-control instrument of the distance form (pilot controller) there might be: (1) Time required for a variation in controlled vessel to reach primary element (metering reaction time); (2) time taken by primary element to actuate pilot valve; and (3) interval from changed position of pilot to changed position in power cylinder; (but the momentary period between motion of power device and effect on fluid flow through the controlled valve comes under application lag)."

The application lag (time lag) arises from the fact that practically every physical system which is subjected to automatic control has a capacity or storage. Consequently, there arises a time interval between the performance of a corrective function by the controller itself and the assumption of the changed condition by the system. Thus, if an added quantity of steam is passed by an automatically controlled pressure-reducing valve in a pipe line, it takes this additional quantity of steam a definite time to travel the length of the pipe to the consumer. The application lag may vary from a microsecond to several minutes, depending upon the application. But irrespective of the magnitude of this time lag, if the application demand requires a changed supply, then the complete system must pass over into the new circumstance with aperiodic fluctuation or vibration of the controlled function about its normal value.

A truly rational solution of a control problem cannot divorce the controller from the application. It must take into account all of the variables which are involved. Such mathematical analyses have been made. In setting up the equations it is, of course, necessary to introduce simplifying assumptions in order

[3] "The Manual of Instrumentation," by M. F. Behar, Instrument Publishing Company, Pittsburgh, Pa., 1932.

[4] Republic Flow Meters Company. Mem. A.S.M.E.

[5] "Governors and the Governing of Prime Movers," by W. Trinks, D. Van Nostrand Company, New York, N. Y., 1919.

[6] "Technische Schwingungslehre," by W. Hort, second edition, Julius Springer, Berlin, Germany, 1910.

[7] "Die Regelung der Kraftmaschinen," by M. Tolle, third edition, Julius Springer, Berlin, Germany, 1909.

[8] "Dampf- und Gasturbinen," by A. Stodola, sixth edition, Julius Springer, Berlin, Germany, 1922.

[9] "Regelung und ausgleich in Dampfanlagen," by T. Stein, Julius Springer, Berlin, Germany, 1926.

to obtain solvable differential equations. This does not necessarily bring in any basic lack of agreement between the actual and mathematical characteristics of the vibration process. The assumptions are made so that linear differential equations will arise; these being the only type that are readily solvable. The essential difference between the actual and the mathematical results is that the latter give the vibration wave a sinusoidal character. The same simplifications are used in studying electrical phenomena, and are even more permissible in controller problems since only a few values are needed quantitatively. The rational analysis has for its object the determination of the following: (1) The conditions for stable control, that is, an aperiodic vibration which dies out in a few regulator swings; (2) the maximum amplitude of the departure from normality; and (3) the period of a control process oscillation, since this must not exceed a practical value.

To go into a series of such analyses would be time consuming, and those interested are referred to the works of Stodola[8] and Stein.[9] However, it is important to bring out the following:

If ϕ is the ratio of the departure from normality of the controlled function to its normal value, the equations which arise will depend on the controller design and on the application; they will, however, be of the following forms:

$$C_0 \frac{d^2\phi}{dt^2} + C_1 \frac{d\phi}{dt} + C_2\phi = 0 \qquad [1]$$

$$C_0 \frac{d^3\phi}{dt^3} + C_1 \frac{d^2\phi}{dt^2} + C_2 \frac{d\phi}{dt} + C_3\phi = 0 \qquad [2]$$

$$C_0 \frac{d^4\phi}{dt^4} + C_1 \frac{d^3\phi}{dt^3} + C_2 \frac{d^2\phi}{dt^2} + C_3 \frac{d\phi}{dt} + C_4\phi = 0 \qquad [3]$$

$$C_0 \frac{d^n\phi}{dt^n} + C_1 \frac{d^{n-1}\phi}{dt^{n-1}} + \ldots + C_n\phi = 0 \qquad [4]$$

The conditions for stability and aperiodic vibration in Equations [1], [2], and [3] are satisfied when (*a*) all factors are positive, $C_1 > 0$; (*b*) when $C_1 - C_2 - C_0C_3 > 0$; and (*c*) when $(C_1C_2 - C_0C_3)C_3 - C_1^2C_4 > 0$.

The C's are constants relating to the application and the controller. The order of the equation for a particular case will depend on the number of capacities and controller elements which are involved.

It should be noted that one of the basic requirements for stability is that the coefficient of the $d\phi/dt$ term shall not vanish. This is a damping term. The stabilization may (1) arise from the reaction of the application on the controller, due to an additional flow of fluid, (2) it may be effected by an oil dashpot located between the controller and the power piston, or (3) it may be obtained by controlling over a "zone" or "band." The ideal is precision control with stability. This can be attained by using a spring, which gives temporary band control, in combination with a follow-up, and a dashpot, which gradually eliminates the effect of the spring; thereby furnishing control to a fixed value of the controlled function.

From the foregoing it is seen that Mr. Mitereff's paper is concerned with a single component, the metering reaction time (time required for a vibration in a controlled vessel to reach the primary element) of the controller period. His particular use of the term "time lag" for what might be called "metering reaction time" is confusing. His lack of differentiation between "controller period" and "application lag" may be responsible for the contradiction contained in paragraph 6, p. 163. It also becomes apparent that Mr. Mitereff has not discussed the "rational approach to the control problem" but that he has attempted to classify controllers on the basis of the metering reaction time. While not always possible, the "metering reaction time" can be made zero. One might erroneously infer that since this would nullify what Mr. Mitereff calls the "time lag," in such a case the fluctuating characteristic of a control change in an application with storage would be eliminated.

If a classification of controllers is desired, the writer prefers one following the plan presented in the excellent article on flow controllers, by Ed S. Smith, Jr.[10]

In closing this discussion, it must be pointed out that the real problem today is to simplify the modern controller without sacrificing its excellent characteristics. A brief inspection of automatically controlled processes will show that they work, and work remarkably well. Whether or not the Class-XII controller illustrated by Mr. Mitereff has any advantages over the best modern controller is problematical but it is obvious that it is more complicated. His paper called to mind the following passage concerning differential equations:[11] "A function that satisfies a differential equation is a solution no matter how obscure its origin, and one that does not satisfy it is not a solution, no matter how illustrious its pedigree may seem to have been."

Ed S. Smith, Jr.[12] Any attempt, such as the author's, to convert an art into a science is likely to be a temporary step backward on account of the oversimplification usually required for setting up basic formulas, as the writer learned when he made an analysis of flow controllers for industrial use[10] which was less formal mathematically than other analyses in this field. Certain of the principles stated by the author as being more or less new are recognized as established practice with governors for steam and water turbines. In one division alone of the U. S. Patent Office are listed the following patents on "Control by Higher Derivatives:" 1,436,280; 1,497,164; 1,703,280; 1,860,821; 1,916,477; 1,936,763; 1,946,280; and 1,955,680. Nevertheless, the author's concise statement of the basic relations in mathematical form is commendable.

Physical and mathematical analyses of controllers for boilers have been published by Hodgson and Ivanoff in England. While there are differences between the author's and the foregoing presentations, these are expected, and even desirable, in the present condition of the literature on controlling. Further, it seems advantageous to have an early symposium at which other viewpoints could be briefly presented. After this airing of the whole matter, those participating in the symposium could form a research committee to consist only of members who, as a small group, would actively agree upon the definitions and basic relations involved for immediate publication. Some material of interest in this direction has been given by M. F. Behar.[3]

In an analysis of control, the phenomena may be taken as transient and subject to cyclical variations. It is then convenient to use some form, possibly series, of cosine expression for the relation between the controlled variable and time. This formulation clearly brings out the phase relations, as of inertia effects, for example, that determine whether hunting oscillations tend to increase or decrease in amplitude, which tendency is expressible by a suitable decrement, as is familiar in alternating-current and vibration-damping theory.[13,14] This treatment also furnishes

[10] "Analysis of Fluid Rate Control Systems," by Ed S. Smith, Jr., *Instruments*, vol. 6, March, 1933, p. 54.

[11] "Advanced Calculus," by W. Osgood, The Macmillan Company, New York, N. Y., 1922.

[12] Hydraulic Engineer, Builders Iron Foundry, Providence, R. I. Mem. A.S.M.E.

[13] "Vibration Damping, Including the Case of Solid Friction," by A. L. Kimball, Trans. A.S.M.E., vol. 51, 1929, paper APM-51-21, pp. 227–236.

[14] "Steady Forced Vibration as Influenced by Damping," by L. S. Jacobsen, Trans. A.S.M.E., vol. 52, 1930, paper APM-52-15, pp. 169–181.

the optimum values of the constants for given control conditions. It is believed that engineers generally make more use of such solutions than of the corresponding differential equations as presented by the author. Since continuity was assumed with the previously discussed differential equations and their solutions, these fail to apply strictly to cyclically operated controllers to which, however, their teachings still apply broadly.

It seems ideal in practice to have a cyclically operated meter responsive to the controlled variable to govern the control cyclically, since such a meter has time to respond to one controlling act before initiating another. Further, such meters flexibly lend themselves to desirable integrating and differentiating modes of control. Also, as to the author's second paragraph, adequate data are seldom available at the time of ordering control equipment. This equipment must be made in quantity commercially and consequently have enough flexibility and ease of adjustment so that it can be readily fitted to the individual control application.

For general use with liquid-level, flow-rate, and temperature controllers, the various electrical means proposed by the author seem too rapid in their action. Thermal, hydraulic, and kinetic means of longer period are generally more suitable for such industrial controllers.

H. A. Rolnick.[15] The rational solution of automatic-control problems has been scarcely attempted, or if it has been attempted, the published results have been meager. The author of this paper is to be congratulated for his study on a rational solution of automatic-control problems. His formulation of the types of control actions points the way to a convenient classification from a fundamental viewpoint which should go a long way toward clearing up the present confusion of terms and ideas in automatic control.

The complete theoretical solution of an automatic-control problem involves a knowledge of the process or condition to be controlled. Furthermore, the complete solution involves the interaction of controlling element and controlled element. How difficult this becomes is well known to all who have tried to find a complete solution. Some idea may be obtained of the difficulty by reading chapter 9 of "Governors and the Governing of Prime Movers," by W. Trinks[5] on comparatively simple problems of speed control.

As a result, the development of automatic-control mechanisms preceded the development of any theory. The simplest kinds of control were originally satisfactory for the simple processes when the high accuracy demanded today was unknown. As industrial processes developed in complexity and as higher and higher standards of accuracy were demanded, the automatic-control mechanisms increased in flexibility and effectiveness. The numbers and types of automatic-control mechanisms have increased so greatly in the last few years that a fundamental classification will clarify the entire problem and help to determine which way to proceed.

While such fundamental considerations should be useful in complex control problems, it is well not to lose sight of the fact that the large majority of automatic-control applications of economic necessity still make use of the on-off or two-position controller, which is the simplest type of control. In these applications the question of sensitiveness is an important one.

Those experienced in automatic control know how important sticking in a control valve may become. The problem of sticking is not unimportant when one stops to consider that control valves and sometimes whole control mechanisms must be subjected to corrosive atmospheres or placed in the open and subjected to climatic conditions.

[15] Physicist, Brown Instrument Company, Philadelphia, Pa.

An illustration may be valuable. In the control of a petroleum-cracking unit the temperature of the oil has to be maintained within a few degrees or $\pm$ 0.5 per cent, so that the total permissible variation in fuel flow to the furnace of the cracking unit is $\pm$1 per cent of the total flow. Although control mechanisms have been designed which will respond to a change of temperature of 0.1 per cent, it is difficult to find a control valve which will respond to a change equivalent to less than 1 per cent of its full travel.

As previously mentioned by the writer, the author's classification of automatic-control apparatus is interesting. Some remarks, however, on the particular classes are in order here. Later in this discussion, some of the difficulties of applying these simple formulas will be described.

Considering the author's Class-I regulators, they give that type of regulation in which the controlling element has to overcome fluid friction. The control produced is stable but slow to act, so that rapid variations in the controlled element are not well taken care of and rapid, wide variations in the controlled temperature, pressure, or controlled element follows wide, rapid variations in supply or demand. In the author's example of the Class-I regulator, Fig. 3 of the paper shows a needle valve 4 in the fluid-supply line. This could hardly produce a rate of flow proportional to the pressure difference across it whether the flow was viscous or turbulent. Since a needle valve is an orifice unless it closely approximates a capillary tube in size and shape, the rate of flow through it will be proportional to the square root of the pressure drop. However, the liquid dashpot will bring about the desired action.

The author's Class-II regulator is the most usual type of control. Most on-off controllers, three-position controllers, and throttling controllers behave according to this general rule. The chief fundamental difference between these controllers is the value of the constant k_1. Its value defines what has been called "throttling range" or "sensitivity." If its value is large so that a small change in the controlled element produces a large valve movement, we get nearly on-off action, i.e., a narrow throttling range or high sensitivity. If the value of k_1 is small, so that a large change in the controlled element is necessary for a large valve movement, then we have a wide throttling range or low sensitivity. In some control mechanisms this constant is adjustable and is adjusted to suit the process.

At first glance one would wonder why it is necessary to diminish the sensitivity of the control apparatus. One must remember, however, that in most control applications there is an appreciable time lag between a control-valve movement produced by a change in the controlled element and its effect on the instrument doing the indicating and controlling. Take, for example, a thermostat placed in a room to control the temperature by turning off the steam to a radiator in the room. As the room comes up to temperature from the cold state, the radiators must be considerably hotter than the room. When the control temperature is reached, the thermostat acts to shut off the steam from the radiator. Since, however, an appreciable length of time must elapse before the radiator cools, the temperature keeps rising for some time until the heat output from the cooling radiator equals the heat lost through the walls of the room. Then the temperature falls, and, when the control temperature is reached again, the steam is admitted to the radiator. Since some time is required before the radiator is warmed up sufficiently to supply the heat lost through the walls of the room, the temperature falls below the control point and the process is repeated. If instead of opening the steam valve completely it is only partially opened, a more gradual action would take place and it would be possible to eliminate constant cycling of the temperature above and below the control point.

The kind of control given by the author's Class-III regulator

is typical of the most successful control instruments in use today. The addition of the integral term produces a valve action which keeps the control point at the desired position regardless of changes in energy supply or demand.

The term including the integral provides what has been called "escapement action," "automatic reset," and "load compensation." Its function is to keep at the control point regardless of changes in energy, demand, or supply.

There are several ways of looking at Equation [3] for the Class-III regulators. Normally the valve opening in a temperature-control application is proportional to the heat supply, so that we may write

$$H = k'P + k'' \int_{t_1}^{t_2} Pdt$$

where P = the temperature of the element controlled, and H = the heat supply. If the second term on the right is temporarily neglected

$$H = k'P \quad \text{or} \quad H/P = k'$$

i.e., the throttling or follow-up action tends to maintain a constant proportionality between the heat supply and temperature. If the heat supply should increase, the temperature would rise, or if the temperature falls due to increased demand, the heat supply would fall; any change in energy demand would give a temperature different from that at the control point.

In practice the constant of proportionality is adjusted so that no hunting action takes place. Therefore, at the control point

$$H_c = k'P_c + k'' \int_{t_1}^{t_2} P_c dt$$

where the subscript c refers to conditions at the control point. Then

$$H - H_c = k'(P - P_c) + k'' \int_{t_1}^{t_2} (P - P_c)dt$$

Since the heat demand is nearly proportional to the temperature, the difference in temperature is proportional to the extra energy required and the integral gives the additional energy supplied, or vice versa. Hence, the valve opening supplies just about what the process demands to keep the control on the line.

In cases when the controlled process has a very long time lag so that the constants of adjustment take care of slow changes, a wide, rapid variation in energy demand or supply will produce momentary deviations of the controlled element from the controlled point. When these rapid variations occur frequently, the type of control given by the author's Class III does not produce as satisfactory a control as is sometimes desired.

The type of control given by Class IV is produced by the addition of a spring to the pressure regulator shown in Fig. 3 in the paper to oppose the action of the pressure on diaphragm 1.

The type of control given by Class V is similar to that in the Class-III regulator with the addition of an element which produces a force dependent on the control-valve position. This additional element, according to the author, provides an automatic adjustment of the primary impulse.

The types of control shown in Classes VII to XII are claimed to be novel. They differ from the first six types by the addition of terms involving the rate of change of the controlled element and the rate of the rate of change of the controlled element. While some of the complex types involving several terms are novel, the Classes VII and IX, shown by the author's Equations [7] and [9], have been claimed before. Since the others are combinations of these two classes with the previous six classes, they can hardly be considered entirely new.

In Class-IX controls, a patent issued to L. Behr, No. 1,497,164 on June 10, 1924, on control method and apparatus, claims a mechanism which produces a control movement proportional to the sum of deviation of the controlled element and the second derivative with respect to time of the controlled element. An examination of this patent indicates that it follows Class VII rather than Class IX.

A modification of the Class-VII control types which follow the law

$$F + k_1(dF/dt) = k_2P + k_3(dP/dt)$$

are described in two patents, one by Guido Wunch, No. 1,920,827 (reissue No. 19,276 granted August 14, 1934), and one by T. R. Harrison, No. 1,946,280, granted February 6, 1934. Both of the mechanisms described in these two patents provide a rate of change control due to the term (dP/dt).

The types of control given in Classes X, XI, and XII make use of a rather formidable array of equipment. This is not to conclude that applications for these may not be found, but it is the writer's belief that the successful control applications which can be made with a rather complicated control mechanism are not great in number.

The justification for types of controls producing valve movements proportional to the rate of change of the controlled element is the elimination of the effect of time lag on the control of the process. Theoretically, this is quite sound and it has been recognized for some time by the anticipatory features of control instruments. Also, the effect of inertial forces in general require control-valve movements dependent on the second derivative with respect to time of the deviation of the controlled element.

Some of the difficulties in applying control mechanisms of Classes VII to XII, inclusive, particularly in temperature control, revolve about the fact that for process conditions which vary widely, the constants in the equations change so that additional adjustments are necessary, so that what was intended to be full automatic control must be hand-adjusted when widely varying conditions are met. For example, the constants in the case of temperature include heat-transfer constants which are not constant but vary among other items with flow rates, temperature, and pressure, and sometimes vary quite widely.

It would be rather foolhardy to prophesy the future of automatic-control development. It is hoped that the field of usefulness of automatic control will widen continuously. The author should be thanked for an instructive classification of types of control which helps clarify the situation from a fundamental standpoint.

Time and experience with the equations and classifications given by the author will determine their ultimate usefulness just as the interaction between the controller and the controlled element brings out the ultimate usefulness of a controlled mechanism.

C. S. Robinson.[16] Mr. Mitereff's paper in its full development should prove of great assistance not only in regulator design but also in the design of the entire system, particularly since the application of any of several excellent control devices now available will have as much bearing on the quality of the result secured as the design of the regulator itself.

In order to follow such changes closely and accurately with the minimum disturbance to the system under control, the regulator should be arranged to receive without distortion the actuating impulses the instant they are generated and should be designed to convert these impulses instantly to a force

[16] Engineering Department, E. I. du Pont de Nemours & Company, Wilmington, Del. Assoc-Mem. A.S.M.E.

sufficient, and only sufficient, to correct the change causing the impulse.

The division of the system into its major parts, that is, (*a*) the system to be controlled, and (*b*) the automatic-control apparatus, is not always an easy problem. As an example, consider Fig. 1 of the paper as a simple automatic-control installation. Normally, the float chamber 3, float 4, valve 5, and the linkage connecting these would be considered the automatic apparatus. Mr. Mitereff's equations in the caption to his Fig. 1, which are intended to show the relationship of the levels in tank 1 and chamber 3, are based, however, on the effect of pilot line 2, a part of the installation not normally considered as part of the automatic-control apparatus. Any apparatus, therefore, designed to solve these equations would have to include the pilot line as part of itself.

While the author has probably made several assumptions, not specifically stated in his paper, as a basis for the equations in the caption of his Fig. 1, there are, in the writer's opinion, other factors to be considered before these equations can be considered universally adaptable. The writer contends that the upper pilot line could have, under certain circumstances, an appreciable effect on the relative levels. Furthermore, if tank 1 is considered a boiler and chamber 3 a boiler-feed regulator, the relative levels would seem to be affected by other factors such as the relative densities and the extent of ebullition of the water.

The design of valve 5 in Fig. 1 of the paper would also affect the result obtained even though the float 4 were placed in tank 1, the ratio of the leverage increased, and the other conditions stated in the latter part of the paper obtained.

The writer is also interested in the development of the equations for various classes of regulators. In developing the example of the Class-I regulator, the author states the relationship between the speed of movement of fluid through valve 4 in Fig. 3 of the paper and the speed of movement of valve 2, but does not appear to connect this latter with the expression dF/dT. A more detailed analysis would seem necessary, particularly when considering the various designs available for valve 2.

Recent experience with several valves similar to the type shown in Fig. 3 of the paper, but without a dashpot, offers a definite example of the effect of the valve design on the operation of the entire regulator. The writer found it possible to obtain an excellent characteristic, that is, the relation between P and valve position, when no steam was flowing. Steam flow, however, changed this considerably and it was possible to obtain the desired result only by redesigning the valve. In the case referred to by the writer, the forces acting on the valve sleeve changed in some unknown relation to the change in steam flow which in turn changed $P_1 - P_2$. No amount of adjustment on valve 4 could correct this situation.

E. T. Johnson.[17] The author has given a thorough mathematical analysis of the problem involved in overcoming the effect of time lag in a regulator. While it is admitted that the methods of solving control problems have been more or less empirical and that more rational methods would be desirable, it is questioned whether it would pay in practice to overcome the mechanical difficulties in making a regulator by applying the author's suggestions with all their complexities.

Inasmuch as hunting in a regulator is caused by overcorrection by the regulator itself, due to the various lags in the system, it is difficult to see how a regulator large and sensitive enough to handle the load variations in the installation without any appreciable shift in the control point could be made to move with such precision as not to have any tendency to overcorrect.

[17] Regulator Company, Chicago, Ill. Mem. A.S.M.E.

The writer cannot agree with the author that sensitiveness and sticking are insignificant factors. It is a well-known practice and has been shown by M. F. Behar[3] that reducing the sensitiveness of a regulator will eliminate hunting at a cost of inability of the regulator to maintain the control point with a change in load. If this latter feature were not important, there would be no necessity for other means to eliminate hunting. Also, sticking will cause sudden great variations in the flow of the controlled medium, which is conducive to hunting and should be eliminated entirely if possible.

With a knowledge of the facts that (1) hunting can be eliminated by reducing the sensitiveness of the regulator and (2) a high degree of sensitiveness in the regulator is required to prevent an appreciable shift in the control point with a change in load, there remains for the instrument designer the only alternative of slowing down the valve movement without reducing the sensitiveness of the regulator.

The writer agrees with Mr. Mitereff's opinion that the retarding of the valve by means of a dashpot can only be excused by a want of a better method. Any method for temporarily slowing down the valve movement should be applied as close to the primary element (such as the thermostatic bulb of a temperature regulator or the float of a liquid-level controller) as possible.

It is known that a capacity in the controlled medium will slow down the valve movement and thereby eliminate hunting. A device applied to the primary element and having characteristics similar to that of a capacity in the controlled medium would, therefore, eliminate hunting of the valve. This device would smooth out the effect of violent changes around the primary element by storing up the excess of energy fed into this element by a departure from the normal plus value and feed this energy back into the primary element on the following departure from the normal minus value. This will reduce the regulator's sensitiveness to sudden fluctuations in load but not to fluctuations of a longer period.

It is well known and has been pointed out by M. F. Behar[3] that a regulator should react faster to a large sudden departure from normality than to a smaller one. This can be accomplished by having the device gradually build up resistance to the storage of more energy.

How a device of this kind is applied to a self-operating temperature regulator is shown in Fig. 1 of this discussion. In this regulator the bulb 5 and the capillary connecting tube 3 are normally filled with a volatile liquid while in the vaporizing chamber 46 where normally exists a partial vacuum. A slight increase in temperature around the bulb 5 will force a portion of the liquid into the chamber 46 where it will flash into vapor and close the valve with a throttling action. A $^3/_4$-in. valve with a bulb $^5/_8$ in. in diameter and 12 in. long will move from open to closed on a variation of 1 F.

In installations with little or no capacity a regulator of this type will hunt due to too rapid valve movement with consequent overcorrections. The hunting eliminating device consists of a shell forming a liquid-filled chamber 79 (this chamber communicates with the rest of the thermostatic bulb 5 through the capillary tube 76, the length of which is a certain predetermined fraction of the overall length of the capillary connecting tube 3 between the valve and the bulb 5), and a flexible metal bellows 77 sealed at both ends and filled with a compressible fluid such as air.

Under normal operating conditions, the pressures on both sides of the bellows 77 are in equilibrium. A decrease in load with consequent increase in the temperature of the controlled fluid will expand the liquid in the thermostatic bulb 5.

Because of the lesser resistance in the tube 76 relative to the

tube 3, the greatest part of the increment in volume of the liquid in the bulb 5, as a result of its expansion, will flow into chamber 79 and compress the bellows 77. A small departure from normality of short duration will be almost entirely absorbed by chamber 79. Should this departure persist, the valve will finally become actuated to the full value of the increment volume in the bulb 5, at which time the pressures on both sides of the bellows 77 again will be in equilibrium.

When the departure from normality increases in value, the amount of liquid flowing into the shock-absorbing chamber 79 and the vaporizing chamber 46 increases correspondingly.

As the bellows 77 is being compressed due to the inflow of liquid into the chamber 79, its resistance to compression builds up due to the natural spring tension in the bellows. The pressure in the chamber 79 builds up accordingly with resultant resistance to inflow of liquid through the tube 76. Therefore, as the departure from normality increases in value, so will the amount of liquid flowing through the tube 3 increase relative to that of the tube 76 with a resultant increase in speed of valve movements.

It has been shown that a departure from the normal plus value will force a portion of the liquid in the bulb 5 into the

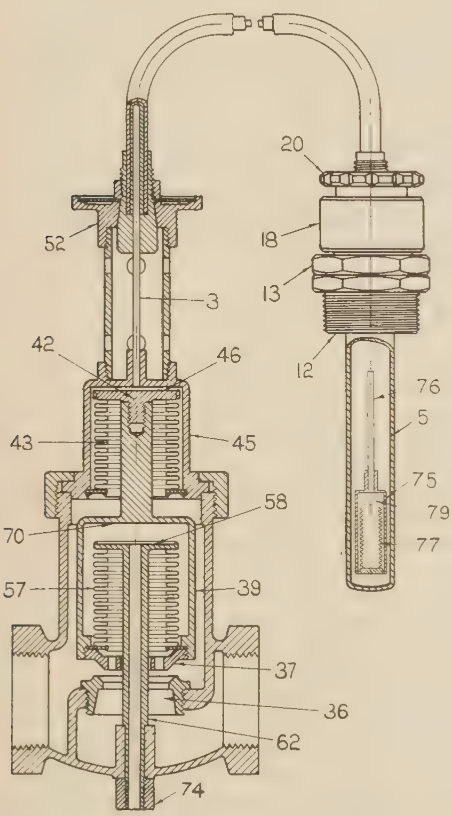

Fig. 1 Regulator and Capacity-Controlling Element

chamber 79, thereby temporarily reducing the speed of the valve movement. It will be seen that a departure from the normal minus value will contract the liquid in the bulb 5 with a resultant decrease in pressure in the bulb. The liquid in the chamber 79 will then flow out into the bulb 5 until the pressures on both sides of the bellows 77 are in equilibrium, in the meantime reducing the rate at which the valve throttle 37 will open.

For more complicated processes it might be desirable to adjust the rate at which the spring tension in the bellows 77 increases, which in turn will change the rate at which the valve movement will be speeded up with an increase in the departure from normality of the controlled process.

This can be accomplished by placing chamber 75 in such a position inside or outside bulb 5 that a spring or other means can be made to resist the compression of the bellows 77. Adjustments are made by changes in the effective length of the spring rather than by changes in the spring tension. It is evident that a damping device of this kind can be applied to other types of regulators than the one shown in Fig. 1 of this discussion.

P. W. Keppler[18] and E. A. Salo.[19] The author has introduced equations for expressing the characteristics of regulators. He has also stressed the desirability of dP/dt as a regulator characteristic for counteracting time lag. These appear noteworthy contributions to the art of automatic control.

This paper concentrates so much on lag in control lines that the dP/dt regulator might appear restricted to this one field. However, many other and more formidable time lags have to be dealt with. It would seem that dP/dt, being a direct measure of difference between input and output, could be used to guide the regulator in any case where great time lags are encountered and very good accuracy is desired.

The writers have used a simple device to regulate the coal input into ball-type unit mills, the response of which was slow. It was desired to have the overtravel of the feeder speed regulator vary with the rapidity of the control demand, which is dP/dt, because full travel was known to be necessary to take care of rapid changes in load and steam pressure. The characteristic equation of this regulator does not contain dP/dt, and is of the general order of the author's Equation [5], though much more complicated. This equation is

$$F = K_1P - [K_2/(K_3 + K_4\int Pdt - K_5\int Fdt)]$$

Nevertheless, if properly designed and adjusted, this regulator resembles the dP/dt regulator so closely in action that a similar regulating effect should be obtainable. The actual regulator contains a servo-motor and a speed-measuring device and is attached to rheostats; it really controls mill output. Its principles may better be illustrated diagrammatically, assuming control of compressed-air pressure, and omitting the servo-motor.

Fig. 2 of this discussion shows at the bottom a compressed-air line with supply from the right. The control valve O regulates the constant pressure N, while the demand at M varies in an irregular manner. The pressure N is led to diaphragm U over which there is a liquid. This liquid communicates through needle valve R with the closed chamber T containing diaphragm V, equal to U in area. These diaphragms actuate valve O by the differential linkage shown. The motion of the diaphragms should here be looked upon as so small that the pressure in T is not affected by it. Upon a change in N, called P in the paper, the regulating effect F is great at first because the pressure in T cannot change immediately, and nearly equals $K \times P \times A$. As the pressure in T gradually changes due to liquid flowing through R, F decreases until F finally equals only $KP(A - B)$. In the actual regulator B equals $0.9A$. The writers have determined the shape of some curves of F against time. In this discussion B was assumed to be equal to 0.95 A which is entirely feasible and brings out the characteristics of this regulator more clearly. The small effect of the liquid level on pressure has been neglected. The liquid flow in the actual regulator is

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[19] Engineering Assistant, United Electric Light & Power Company, New York, N. Y., Jun. A.S.M.E.

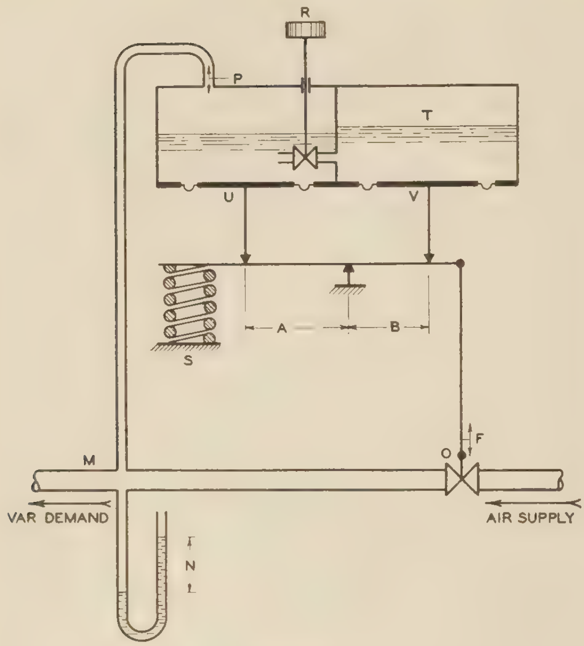

Fig. 2

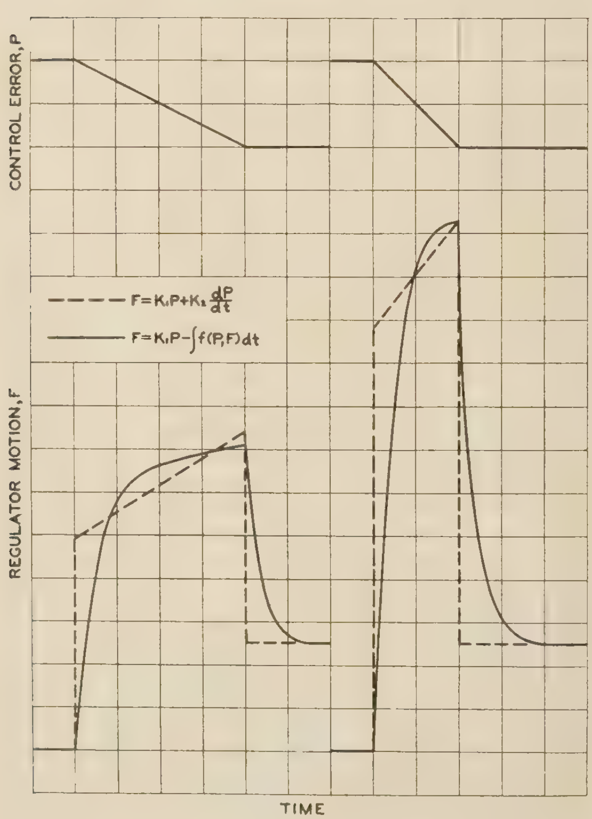

Fig. 3

probably not strictly viscous but a mixture of hydraulic and viscous. This improves its action somewhat, but straight viscous flow was here assumed for the sake of simplicity.

At the top of Fig. 3 of this discussion are shown assumed curves of control deviation P against time. These curves are constant at first, then they fall at a uniform rate, finally remaining constant again. It is convenient to use a simple curve of P. But the sudden changes in dP/dt, as P abruptly starts and stops falling, imply sudden rises and drops in control demand. The dashed curves show the author's regulator with F equals $K_1P + K_2dP/dt$. The full-line curves show F of the regulator shown in Fig. 2 of this discussion. It is seen that the action of this regulator resembles that of the dP/dt regulator quite closely. It is seen that its F rises rapidly as dP/dt changes from 0 to its rate of drop. When P stops falling, F drops off sharply.

To show that this overtraveling effect is approximately proportional to dP/dt, F curves are shown at the right of Fig. 3 of this discussion for double dP/dt, i.e., P falling twice as rapidly. It is seen that F for Fig. 2 of this discussion (full line) is approximately proportional to dP/dt, as seen by the F of the dP/dt regulator (dashed line).

Regulators of this type have given satisfactory service in reducing time lag of unit mills. It might be supposed that it is difficult to maintain the dead air space in the closed chamber T. But this has been giving no trouble whatever. This type of regulator may also be varied in many ways. One interesting possibility would be to replace the top of chamber T with a spring-loaded diaphragm with or without air under it. Without air, the author's Equation [5] is fulfilled. The regulator action, however, does not come as close to the dP/dt type.

In other difficult cases comparatively simple regulators with equation $F = K_1\int Pdt + K_2p$ have given good service. It has been found that K_2 can usually be made quite large without hunting, so that the regulator can be made to overtravel considerably, thus speeding up sluggish equipment.

Nevertheless, for many cases the dP/dt regulator would appear well justified in spite of its inherent complication, because dP/dt is a direct and accurate measure of how far the regulator should move.

R. L. Goetzenberger.[20] In his attempt to classify continuous-type automatic controllers on the basis of the deviation-reaction characteristic, Mr. Mitereff has made a thorough and commendable mathematical study. However, it seems to me to lack a great deal in being a treatment of the principles of automatic control as the scope of its title might indicate.

The fact that little may have been published on the subject of automatic regulators does not imply that manufacturers of this apparatus have not explored scientific grounds and are not familiar with the fundamental equations expressing characteristics. Where empirical methods have been resorted to in the solution of any problem it is because the physical conditions under which the apparatus is to operate and the rare demands for extreme precision do not warrant the utilization of complicated mechanisms with consequent high apparatus costs. However, a great deal of thought is being given to the development of devices that will more nearly solve automatically the mathematical equations and still keep them within the realm of practical and economical application where performance, accuracy, and low maintenance expense must be preserved. The trend of this thought has been recorded ably by M. F. Behar,[3] who not only gives industry and the instrument maker an up-to-date treatise but also educates them in the use of a common language through suggested terminology, a contribution worthy of serious consideration by standardization committees of the engineering societies.

Certainly no control manufacturer is desirous of conducting experiments at his customers' expense, yet it must be recognized

[20] Manager Industrial Regulator Division, Minneapolis-Honeywell Regulator Company, Minneapolis, Minn. Mem. A.S.M.E.

that the complete solution of an automatic-control problem involves more than a concept of the characteristics of a regulator which can be expressed mathematically. It also necessitates knowledge of the wide variations in processing conditions. Therefore, it is not unreasonable to expect that in the instance of unusual installations some cut-and-try methods of adjustment will be required, and this may even exist in the instance of the more general ones until there have been accumulated constants for all typical examples which, in the case of temperature, include those for heat transfer that are not constant but vary with such factors as temperature, pressure, rates of flow. Experience, gained through well-engineered applications, coupled with the equations advanced by Mr. Mitereff seems to offer the practical solution to most automatic-control problems.

Arthur Edwards.[21] Mr. Mitereff's paper is a notable contribution to the subject of automatic control. The formulas presented are distinct and apply to a certain aspect of the system. One would have to apply them to difficult cases in order to evaluate them and comment on their use. A number of control factors such as the speed-load characteristics of the machines governed have been eliminated purposely from the formulas and an attempt to include them would only complicate a paper which already has had to be cut for publication. It is to be hoped that Mr. Mitereff will cover these related phases of regulation in his forthcoming book.

Unfortunately, Mr. Mitereff will discover that a large proportion of the regulator-using public is frightened more by an integral sign than by the fact that regulation has been haphazard in the past. This paper is of value, however, to persons who are responsible for the development of the art, and the average user need not be aware that a differential equation lies behind the development.

E. F. Hanford.[22] Mr. Mitereff has presented a qualitative mathematical analysis of the behavior of automatic-control apparatus which merits considerable study. In designing a new plant the engineer selects the type of control which his experience tells him is best fitted for the problem at hand. He may visit the plant a year or so later and find the automatic control disconnected or inoperative and wonder at the stupidity of the operators. The engineer and operator have both done their best, but the operator has come to the conclusion that while the controls can be made to work, he can get better performance without them.

After the plant is in operation, plant engineers are careful in installing automatic controls and rarely do so unless they have had some experience with an identical installation. They have to talk to the management in dollars and cents.

These same engineers would have installed almost any other piece of equipment with less hesitation. The difference is due to the fact that the characteristic behavior of the several types of controls is only known empirically. The past twenty years have seen an ever-increasing invasion of these robots, good and bad. With the assistance of Mr. Mitereff's dissertation on automatic-control problems, engineers will probably learn that all automatic-control equipment will operate successfully when properly chosen.

Endres Zihlas.[23] Mr. Mitereff's paper is of interest and deserves commendation. The lag element in automatic-control apparatus as explained by the author is an important feature of such equipment. With the aid of the equations derived in the paper, the accuracy of automatic regulating mechanisms should be increased and undue waste of fluctuating control eliminated.

W. F. Ryan.[24] There is much food for thought in Mr. Mitereff's paper for the designer of any important control. It is of little importance to any one, except perhaps Mr. Mitereff, whether or not the Class-VII to Class-XII regulators are really new. The important questions are:

1 Can the desired results be obtained more directly, more positively, more simply, or more cheaply than they are commonly obtained by existing devices?

2 Can the painful cut-and-try period, which is usually experienced on any really new control problem, be shortened by a *rational* analysis of the problem?

There has been much valid criticism of Mr. Mitereff's paper, but some of the discussion might have been omitted if the discussers had understood the meaning of the word "rational," as it is used in the title. Nevertheless a great mass of practical experience has been built up by cut-and-try and rule-of-thumb methods, and this experience is of much value in the solution of any problem of regulation.

In spite of the literature which has been cited, consideration of so highly developed a device as the governing mechanism on a large turbo-generator, in the light of Mr. Mitereff's analysis, leads to the belief that the *rational* approach has been somewhat neglected. Differential effects have been recognized in governor design since the days of Watt, but they have been suppressed by great weight and power. Suppose, instead of suppressing these effects, they were turned to account in promoting regulation, as Mr. Mitereff suggests? Would regulation be better, simpler, or less costly? One of the greatest problems in the design of the Panama Canal was solved when the engineers ceased trying to divert the Chagres River and decided to make use of its waters in the operation of the canal.

There is no discrepancy between theory and practice, in any field whatsoever, if the theory is sound and complete. It is the particular function of a society like ours to reconcile theory and practice; in other words, to bridge the gap between the pure scientist and the practical plumber. We must apply the test of practical experience to all new theories before they are accepted. Conversely, we should apply mathematical and scientific analysis to those arts which have been developed chiefly by practical experience.

Mr. Mitereff's paper is an effort to apply mathematics and science to the problem of control. He is not the first to make this attempt, but he is perhaps responsible for awakening many members of this Society to its possibilities. In spite of defects which have been pointed out by competent critics, the paper can be made an instrument of much value to designers of regulating equipment if it induces them to attack their problems along *rational* lines, whether they adopt Mr. Mitereff's analysis, or apply original methods of their own.

Author's Closure

P. S. Dickey remarks that the paper omits the equation expressing the relationship between the motion of the final operating element and the rate of flow of energy of a fluid to or from the storage. For dynamic conditions the form of this equation will depend upon what M. F. Behar[3] calls "application time lag." It was taken for granted, moreover, that the ports of a regulating valve should be so designed as to result, at static conditions, in the rate of flow substantially proportional to its opening.

[21] Boston, Mass. Jun. A.S.M.E.

[22] Lever Brothers Company, Hammond, Ind.

[23] Oswego, N. Y.

[24] Stone & Webster Engineering Corp., Boston, Mass. Mem. A.S.M.E.

By an electrical contactor was meant a rheostat which is an electric counterpart of the fluid resistance of a valve.

A pilot line was selected as a typical example of the time lag simply on the basis of the case of analysis. However, in the complete text of the paper[25] an analysis is made of the time lag which occurs often in the response of the system to the action of the regulator ("application lag").

Attention is called in this connection to the statement in the paper that "The time lag arising from the use of a pilot line is typical of any other source of time lag in any other part of the controlled system." Due to lack of space for detailed explanation, it was intended that this statement should be taken at its full face value. This latter kind of time lag is probably more important from a practical standpoint than is the time lag in the transmission of the primary impulses, as P. W. Keppler and E. A. Salo pointed out in the example from their experience.

Another example of the time lag between the action of the regulator and the response of the system to it was cited in current literature.

An installation was described in which it was desired to control the speed of induced-draft fans driven through a hydraulic type of variable reduction gear, the time lag occurring in the gear. After some experimentation, the problem of encountered violent "hunting" was solved by the installation of auxiliary dampers. A much more economical and satisfactory solution could have been obtained by the use of a regulator of characteristic VII or perhaps IX, depending upon the particular kind of time lag of the reduction gear. Such rational solution would have eliminated the waste of power due to auxiliary dampers of the empirical solution adopted.

The voltage regulation of a generator is another good instance where characteristic VII would be of great advantage in counteracting the lag due to electrical inertia of the magnetic field.

Referring to P. R. Dickey's criticism of the solution of the level-control installation shown in the paper, the solution presented is quite correct and complete. For the installation shown in Fig. 1 of the paper, for instance a regulator of characteristic IX should be used to (1) counteract the resistance of the pilot line 2 if it were very short and (2) counteract the inertia of fluid in line 6 if it were comparatively long. In this case the constant K_1 in Equation [9] could be increased so greatly that the shape of the valve 5 would be of practically no importance; even a disk-shaped valve would be sufficient.

In so far as the three-element type of boiler-feedwater regulator is concerned, the author fully recognizes its advantages in this particular application as well as in all other installations where it is either impractical or impossible to measure accurately the amount of fluid or power in storage. This however, does not occur very often.

On the other hand, if it were practicable to suspend a boiler on a scale, so as to be able to measure the amount of water in it independently of "swelling," a regulator of simple characteristic II would have given even better results.

The inconsistency in the use of symbols noted by P. S. Dickey is only apparent, since the distance traversed by a regulating valve *is* the final regulating effect of a regulator actuating the valve, while the rate of flow through the valve depends upon the conditions of the system external to the regulator.

The discussion of M. J. Zucrow is not very pertinent. The term "a measuring instrument" is not a particularly good one, since there is nothing about the diaphragm of a pressure regulator, for instance, which would indicate the pressure applied to it, especially if there is no relationship between the pressure and the distance traversed by the diaphragm. The author could never understand the reason for voluminous theoretical literature on speed governors of steam turbines and engines. Due to an almost complete absence of the time lag all one has to do to completely solve the problem is to design a very accurate and powerful governor having the characteristic II.

That the practical difficulties of designing such a governor are not insurmountable is attested by an excellent fluid-pressure governor of this characteristic developed in recent years by the Westinghouse Electric and Manufacturing Company.

The division of regulators into self-operating and relay controllers is unimportant, since both of these types can have practically any basic characteristics. If a great power is required a relay would naturally be used.

The term "controller period" mentioned by Dr. Zucrow is just another name for the time lag inside the regulator itself. This time lag distorts the performance of a regulator as compared with its theoretical basic characteristic and it is the duty of the designer to minimize this distortion. The first step, however, is to decide just which basic characteristic is correct for a given system external to the regulator, because if the basic characteristic is not properly selected, even a perfect regulator of this characteristic will be unsatisfactory.

The classification in the paper is based not on "metered reaction time" but is made according to the basic performance characteristics, that is, according to the fundamental relationship between the primary impulse actuating the regulator and its final regulating effect (movement of the valve).

The author is in perfect agreement with the quotation closing the discussion of M. J. Zucrow.

The classification given by Ed S. Smith, Jr., is incomplete and his designation of the corresponding classes of the paper is purely verbal and therefore indefinite. One may call a regulator of Class II "corresponding" or a regulator of Class III "compensating" but it is much more definite to express its performance by a mathematical equation.

The examples of the regulators in the paper are mostly illustrative, the commercial designs of the rate-of-change responsive regulators are covered by author's U. S. Patents Nos. 1,955,680; 2,015,861; 2,015,862; 2,020,847; 2,022,818.

The author is in accord with the suggestions of Ed S. Smith, Jr. that a standardization committee, preferably under the auspices of A.S.M.E. should be set up to decide on the terminology, classification, and the rules of selection for automatic regulators, as well as on the procedure of acceptance tests.

Such terms as "sensitiveness" versus "sensitivity" are very confusing at present.

"Sensitiveness" can be defined as the degree of accuracy of response to primary impulses of small magnitude, whereas "sensitivity" is nothing else but the value of constant K_1 in Equations [2], [7], and [9] of the paper.

The author used in his studies a sine function as the basis for graphical investigation of the problem of "hunting" with gratifying results.

The possibility of practical design of a cyclical regulator involving the differentiation by the step-by-step method is recognized by the author and he hopes the manufacturers of such regulators will not be slow in putting this development on the market.

The field of temperature control is in great need of this development. In the installations where the rate of heat transfer varies considerably, an automatic adjustment of the constant coefficients in accordance with the load will be advantageous.

H. A. Rolnick made an excellent analysis of the regulators of the first five classes.

Class IV is really the Class I provided with an automatic

[25] The author's published paper did not include a complete demonstration of all the regulators mentioned by the author. However, a complete demonstration of all the cases has been filed by the author in the archives of the Society.

adjustment of the primary impulse since Equation [4] of the paper could be rewritten as $k_2F = k_1\int[P - (1/k_1)F] + c$. Class I is inherently hunting even in the absence of the time lag and should never be used.

In spite of the popularity of Class-III regulator this class is in the nature of an artifice ameliorating but not solving the problem of the time lag, it is, however, the best characteristic now on the market with the possible exception of Class V.

H. A. Rolnick is quite correct in stating that patent No. 1,497,164 shows a cyclical controller of the characteristic VII, (or more exactly of characteristic VIII).

However, the equation $F + k_1(dF/dT) = k_2P + k_3(dP/dT)$ can also be considered as an alternate expression of the characteristic V obtained by differentation of Equation [5] of the paper.

C. S. Robinson's remark that it is not always easy to divide the system into automatic-control apparatus proper and the system to be controlled, is not substantiated by a valid example. It could be stated in this connection, that the equipment shipped by the control manufacturer is an automatic-control apparatus, whereas the purchaser's plant is the system external to the regulator.

The explanation of the characteristic I, cannot be improved by the author as suggested by Mr. Robinson. The regulating valve should be so designed as not to overtax the available power of the regulator, since a sticky or unbalanced valve can not only distort the basic characteristic of the regulator, but can even stop the regulator altogether, if the regulator is too weak for this task. Considerations such as these have very little to do, however, with the basic aspects of the problem investigated in the paper.

The author cannot agree with E. T. Johnson that the "hunting" can be completely solved, without detrimental results, by the introduction of an artificial time lag, provided this time lag is introduced at the point of generation of the primary impulse. The effect of time lag is substantially the same irrespective of the place of its introduction.

The discussion by P. W. Keppler and E. A. Salo is valuable and contributes to a better understanding of the main thesis of the paper.

E. F. Hanford is quite correct in his analysis of the only too often encountered difficulty in keeping automatic controls in operation. Part of the difficulty mentioned by Mr. Hanford lies in the regulators, since the usual acceptance tests are not at all indicative of the regulator's performance under all and sundry conditions, and if a regulator periodically upsets the operation, the operator is justified in his choice of preferring to watch only the operation rather than to watch both the operation and the regulator.

W. F. Ryan's reference to inertia governors is quite timely.

The characteristic of an inertia governor as previously designed (that is involving an appreciable travel of the weights in radial and circumferential direction in relation to rotation) is substantially

$$F + k_3\frac{d^2F}{dT^2} = k_1P + k_2\frac{dP}{dT}$$

The term $k_3(d^2F/dT^2)$ is due to the retarding effect of inertia of the weights during their travel and it cannot be eliminated in a conventional design. Moreover, even in pure form, the characteristic VII is only detrimental to proper speed regulation of steam-driven equipment due to absence of time lag in such systems.

Whatever success the inertia governors achieved in speed governing of steam engines is due entirely to the fact that weights are damped with a dashpot obtaining the characteristic

$$F + k_2\frac{dF}{dT} + k_3\frac{d^2F}{dT^2} = k_1 + k_2\frac{dP}{dT}$$

If k_3 is made so small as to practically eliminate term $k_3\,(d^2F/dT^2)$, this characteristic reduces with proper adjustment to Equation [2] or $F = k_1P$ which is quite correct in this application as was pointed out before.

Incidentally the discoverer of the rate of change responsive characteristics is Nicolai Minorsky (U. S. Patent No. 1,436,280) who gave a rational solution of the problem of the steering of ships by the application of these characteristics. There is no evidence, however, that at that time Dr. Minorsky realized all the implications of his discovery.

The author wishes to acknowledge his indebtedness for invaluable help and encouragement given to him by W. F. Ryan in preparation of the paper.

Draft-Gear Action in Long Trains[1]

L. K. Sillcox.[2] The problem before the manufacturers of railway draft gears has been constantly that of installing, within a specified and definitely limited space, elements capable of presenting the maximum energy-absorption capacity consistent with tolerable end loading transmitted to the car structure. It is no longer a question of how much capacity is required under service conditions—the living necessity is for the utmost capacity which can be crowded into the standard pocket. No matter how rapidly capacity has been increased, or how far technical ingenuity has permitted the introduction of gear types offering this capacity with attendant and suitable characteristics of release, sturdiness, and durability, the demand has been such as to eclipse the ability of the perfected gear to protect positively the car structure against damaging shocks in view of modern operating conditions.

While the need for ultra high energy-absorption-capacity draft gears has become particularly urgent to relieve properly the car structure of stresses which it would otherwise be called upon to withstand, notably under conditions of buffing shocks, instances have developed when this high capacity, obtained by raising the line of delivered force throughout the range of gear travel, has proved a disadvantage in itself. It has been the cause of hard-riding qualities on the part of modern locomotives during the acceleration period since the maximum tractive effort of the locomotive has, in some instances, resulted in but slight gear movement with an inadequate measure of resilience to cushion the pulsating forces delivered by the pistons. There is then the need for entirely different draft-gear characteristics obtaining at the rear of the tenders of even the most powerful freight locomotives. So it is that capacity alone does not represent the only feature which must be respected in fitting modern draft gears to modern train equipment.

Mr. Wikander's analysis of the draft-gear problem represents a highly commendable endeavor to arrive at the conditions which must be met in order that the draft gear may perform its fundamental purpose. Treated conversely, the railways might benefit by an analysis of the most severe conditions which may be safely imposed upon available gears. The expeditious movement of freight represents efficient handling until the point is reached where the cost of handling exceeds the advantages derived. There is an optimum switching speed which, when respected, will result in the greatest net benefit. But switching speeds are the most readily calculable part of any investigation of train

[1] Published as paper RR-57-1, by O. R. Wikander, in the August, 1935, issue of the A.S.M.E. Transactions.

[2] Vice-President, The New York Air Brake Company, Watertown, N. Y. Mem. A.S.M.E.

shock. The many and unpredictable variables which are encountered in train service are indicated by Mr. Wikander as he establishes the basis of his investigations. To the effects of free slack, distribution of loads and empties, and brake propagation time, practical operation adds factors of braking ratios, a coefficient of brake-shoe friction, which is variable and sensitive to brake-shoe pressure differentials, properly fitting and poorly fitting brake shoes, grade, curvature, and many other disturbing elements with which no analytical survey can deal completely.

The air brake and the draft gear are closely related and essential allies in permitting the safe control of long trains. The air-brake type, although it will, without regard to the manner of operation, stop the train in the shortest possible distance, would be as ineffective an agent in safely controlling coupled cars as would the draft gear selected merely to protect trailing cars in the event of collision. The function of the air brake is that of bringing a moving train to a stop under emergency conditions in the shortest distance consistent with the production of resultant stresses between cars well within the cushioning capacity of the draft gears.

We have witnessed wonderful and simultaneous development of draft-gear and air-brake appliances to the extent that the worst of railway offenders, the long tonnage freight train, traveling at modern speeds, no longer need represent the hazard that marked its introduction. There remains the necessity for the development and enforcement of proper standards of maintenance for the draft gear on the one hand, and proper instruction in the judicious use of the air brake on the other. While the latter need is generally well met and expert manipulation is the rule, draft-gear maintenance is frequently neglected to the extent that the superior qualities which the manufacturer builds into his equipment are permitted to deteriorate and conditions result for which no braking appliance can, of itself, compensate.

Author's Closure

Mr. Sillcox's views on draft-gear problems are of great value because he approaches such problems, not only as a practical railroad man, but also as a designer of air brakes and a trained scientific investigator.

He rightly points out that the basic draft-gear problem is to provide a device which, within the limited space available, will give high capacity without excessive reaction on the car sills.

This, as he says, requires proper proportioning of the gear. Resistance to the initial force must be adequate so that the average resistance may be high without producing an unduly high final car-sill reaction. At the same time the spring action must also be adequate.

Mr. Sillcox calls attention to the fact that high capacity incorrectly obtained may result in the transmission of undesirable vibrations from the locomotive to the train. This is generally due to too great a portion of the capacity of the gear being obtained by friction rather than spring action.

Under the pulsating tractive effort of the locomotive the coefficient of friction drops, the gear creeps shut, and is thus ineffective in cushioning the pulsations. This is a characteristic of gears showing too low a recoil value under the drop test.

If the spring action of a gear is high compared with the pulsating draw-bar pull the hard riding to which Mr. Sillcox refers will be eliminated. To be satisfactory in this respect the draft gear must show a moderately high recoil in the drop test.

Mr. Sillcox's plea for proper draft-gear inspection and maintenance is most timely. The Association of American Railroads has provided specifications for the purchase of draft gears. Proper standards of maintenance are necessary if effective service is to be obtained from the gears.

Rolling-in of Boiler Tubes[1]

L. Skog.[2] The elongation method of tube rolling described by the authors appears to have advantages over the old methods in that some of the guess work is, no doubt, removed in measuring the elongation of the tube during the rolling-in process. It appears, however, that this method would be considerably more expensive, in that an additional man would be required to take care of the indicator gage and also because each row of tubes in a boiler of bent-tube design will have to be set in position and rolled-in before the succeeding rows of tubes can be installed, thereby increasing considerably, the time required and the cost.

The variation in diameter of tube holes given in the authors' Table 1, is much greater than we have found in our practice and must be unusual cases. This variation should not exceed 0.005 in. between the maximum and minimum, which would give ample tolerance for manufacture. Variation in outside tube diameter, which should not exceed $^1/_{32}$ in., is also greater than we have found. These variations in diameter of tube holes and outside diameter of tubes are of importance in getting a satisfactory rolled-in job, no matter what method of rolling is used. It is, of course also important that the outside diameter of the tube be a true circle so that the tube wall will bear uniformly against the tube hole during the rolling-in operation. If the tube is not circular it may touch the tube holes in only one or a few places at which time the indicator-gage needle will be quiet and the prescribed elongation may not give the true condition of the roll. With the elongation method it would be of utmost importance to determine the correct elongation for a particular installation before the job is started because, if the prescribed elongation should happen to be incorrect, all tubes in the boiler may be overrolled or underrolled, this would not be determined before the boiler is tested hydrostatically, in which case all the tubes might have to be rerolled without the aid of the indicator gage.

In most of our installations we have been using the uniform-expander-entrance method, which has given satisfactory results. In the last 16 large boilers installed under the supervision of the firm the writer is associated with, only a few minor leaks appeared during the preliminary hydrostatic test, and after these leaks were tightened by slight rerolling, the boilers were passed by the insurance inspectors.

E. W. O'Brien.[3] Irrespective of the results that may be obtained from the use of the elongation method of tube rolling, the authors have rendered a service to the profession by (1) pointing out the scarcity of information on true tube rolling; (2) analyzing the phases of tube rolling and setting forth the requirements for forming an ideal joint; and (3) bringing to light flaws in the several conventional methods of producing joints, thereby stressing the need for a tube-rolling method that will consistently produce uniformly rolled joints in a given bank of tubes.

The results obtained on the test specimens, the joint surfaces of which were uniformly smoothed and finished, indicate the correctness of the thesis that a definite (though changing) relation exists between tube elongation and joint holding strength. When this relation has been worked out to give the proper elongation for the greatest joint strength in the several conventional joints now employed, it will not be a difficult matter to put the authors' method into general practice.

The writer would be interested in knowing if, when rolling a single tube into a pair of tube sheets at a time when all the other tubes are rolled in place and when for all practical purposes the

[1] Published as paper FSP-57-7, by F. F. Fisher and E. T. Cope, in the May, 1935, issue of the A.S.M.E. Transactions.

[2] Engineer, Sargent & Lundy, Inc., Engineers, Chicago, Ill.

[3] Editor, *Southern Power Journal*, Atlanta, Ga. Mem. A.S.M.E.

sheets are immovable, will the rolling of the tube to a change in wall thickness, equal to that undergone by the other tubes, produce an elongation of magnitude equal to that produced by the other tubes? In this instance, the tube is subjected to compression, and it would seem that rather than producing an elongation, the flow of metal out of the joint would result in a compression of the metal in the remaining portion of the tube; at least, any elongation that would be produced would not be so great as would be produced were the other end of the tube free.

C. H. Fellows.[4] The paper by Messrs. Fisher and Cope has served to call attention to the source of at least one type of boiler-tube corrosion, i.e., pitting immediately outside the tube sheet. The stresses set up in the metal at this point as the result of improper rolling, which generally means more or less severe over-rolling, makes the metal readily susceptible to corrosion, particularly in boilers where untreated or mildly treated evaporated make-up is used.

Corrosion of this particular type is not infrequent in boilers, and although in many instances the condition of the metal has been considered a contributing factor, nothing much could be done about it because there has been no practicable method of rolling-in tubes that would not leave this particular spot in the tube in an abnormal highly stressed condition. Cracked surfaces of the tubes resulting from severe rolling is a condition that probably exists in many tubes now installed and which may be severely pitted as the result of localized concentrated attack of oxygen on highly stressed metal.

The ever present interest in the phenomenon of caustic embrittlement is made more prominent by this paper. When it is considered that abnormal stress is one of the factors which causes this type of embrittlement, it is conceivable that rolling-in boiler tubes as described by Messrs. Fisher and Cope, may limit such embrittlement by reducing to a marked degree the stresses induced in the drum sheet. Although caustic embrittlement in tubes or the sheet adjacent to the tubes is less prevalent than in the vicinity of the rivets, it has been known to occur. Any steps that can be taken to reduce or eliminate one of the factors known to contribute to the development of this phenomenon should be welcomed and carefully investigated by the industry.

Wladimir Noeff.[5] The method of rolling-in boiler tubes described by the authors enables one to control individually the quality of every joint in a boiler, and since the writer has had some experience in this field he takes this opportunity to advance some observations concerning the use of this method in practice.

1 The authors have not pointed out how (*a*) the thickness of the tube wall, (*b*) the decrease in the thickness of the tube wall after it has been rolled, (*c*) the width of the inrolled ring of the tube in the tube sheet, is related to the elongation of the tube. Therefore, they recommend for all cases and all types of boilers the same elongation of 0.02 in. The writer is of the opinion that it will be more suitable to fix the desired elongation to correspond to the variations of the thickness of the tube wall, the decrease in the thickness of the tube wall after it has been rolled and the width of the inrolled ring in the tube sheet. The relation between these quantities in a slightly simplified form can be deduced from a geometrical consideration, Fig. 1 of this discussion, neglecting the increase of the specific volume of the metal by cold working, and also that the form of the tube end and of the tube hole in practice cannot be perfectly regular. In the writer's Fig. 1, the longitudinal section of the inrolled ring of the tube before the rolling is represented by full lines while the dotted lines represent the same section after rolling. In Fig. 1, l is the width of the tube ring before rolling, λ is the axial elongation in one direction presumably equal to one-half of the total elongation,

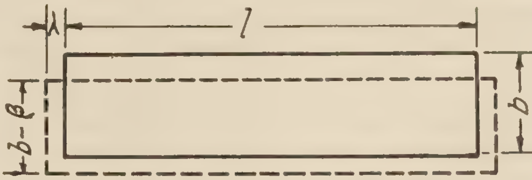

Fig. 1 Diagrammatic Sketch of a Longitudinal Section of Boiler Tube Before and After Rolling

b is the initial thickness of the tube wall, and β is the thinning (decrease of thickness) of the tube wall. According to the writer's reasoning, the areas of both rectangles must be equal, or

$$(l - 2\lambda)\,(b - \beta) = bl$$

or

$$bl + 2\,b\lambda - \beta l - 2\,\beta\lambda = bl$$

Therefore

$$\lambda = \frac{\beta l}{2(b - \beta)} \quad \text{[1]}$$

This formula shows that the controlled elongation at β = constant must be proportional to the initial width of the inrolled tube ring and almost inversely proportional to the thickness of the wall. Therefore, the desired elongation for each type of tube can be determined by using formula [1] of this discussion, once the decrease in thickness of the tube wall has been found experimentally. In reality this was done by the authors when they determined an elongation of 0.025 in. for the tubes in the Ford Motor Company station which operates at 1435 lb per sq in., and an elongation of 0.016 in. for the tubes in the Springwells plant which operates at 400 lb per sq in., apparently because the thickness of the shell walls is greater at the former station.

2 The authors are right when they recommend the use of the elongation of the tubes as an indication of the quality of the joints of rolled-in boiler tubes, but their method of determining the desired elongation can hardly be taken as being correct. In the writer's opinion, the desired elongation must be determined from the thinning of the tube wall which in turn must be determined in accordance with the degree of tube-hole expansion.

3 From the physical point of view, the strength and tightness of the joint are determined, as it appears from the investigations[6] of A. Thum and R. Jantscha, and others, by the degree of tube-hole expansion. This expansion is expressed by

$$\Delta = [(D' - D)/D]\ 100$$

where D is the diameter of the tube hole before rolling, D' is the diameter of the tube hole after rolling, and Δ is the degree of tube-hole expansion. All the tests that are conducted for investigating the properties of the joints must be based either on the degree of tube-hole expansion Δ, or on the absolute increase of the tube-hole diameter or radius as mentioned in the paper[6] by A. Thum and R. Jantscha. It must be assumed that (*a*) the thinning of the tube walls, (*b*) the properties of the tube and shell metals, (*c*) the type of expanding tool, (*d*) the method of rolling-in the tubes,

[4] Chemist, Research Department, Detroit Edison Company, Detroit, Mich.

[5] Senior Engineer of State Consulting Bureau for Steam Power Plants (Orgres) and Editor of *Heat and Power*, Moscow, U.S.S.R.

[6] "Einwalzen und Einpressen von Kessel- und Überhitzerrohven bei Verwendung verschiedener Werkstoffe," by A. Thum and R. Jantscha, vol. 11, no. 12, December, 1930, pp. 397–401.

and (e) the finished condition of the surfaces of the tube and shell which are forced into contact, are all given. It must also be assumed that these factors must be dependent on the degree to which the tube hole is expanded as determined from tests. Knowing these things, one will be able to determine the corresponding thinning of the tube wall and consequently the desired thinning of the tube wall by applying formula [1] of this discussion.

4 It is the writer's opinion that the authors of the paper have not paid sufficient attention to the properties of the tube and sheet metals, particularly when they give the results of their tests of forcing the tubes from the joints. These properties are of the greatest importance in that they determine the best degree of tube-hole expansion, as can be ascertained from Fig. 2, reprinted from the previously mentioned paper[6] by A. Thum and R. Jantscha.

5 That part of the paper "Effect of Lengthening the Tubes Caused by Rolling-In," page 151, does not seem to the writer to be entirely correct, and further, it does not seem to agree with the first part of the paper. Indeed, if we have a method that secures the equality of tube elongation we have no reason to be

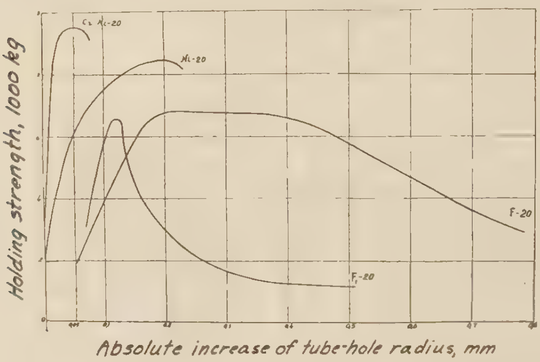

Fig. 2 Holding Strength of Boiler Tubes 3.2677 In. Diameter Rolled in Plates of Different Metals

afraid of any considerable thrust caused by the inequality of this elongation. Also, it is not clear to the writer how the indicator, which is attached to the gas side of the tubes, can be used when all the tubes are inserted previously to rolling-in. It seems that an indicator of this type can be used only in the case where the tubes are inserted and rolled-in row after row.

Also, the author's deductions concerning the thrusts created by unequal rolling cannot be accepted completely. The thrust created by rolled-in tubes surely cannot be added arithmetically, inasmuch as the dividing of the tube bank into two parts does not divide the amount of accumulated thrust in half. All relations that must be considered in this connection are undoubtedly more complicated.

6 In conclusion it must be noted that all the authors' data are obtained from hydrostatic tests after initial tube installations. The writer would be interested in the answers to the following questions:

(a) Have the joints rolled-in by the elongation method proved entirely satisfactory in service?

(b) Have cracks appeared in the joints rolled-in by the authors' method?

(c) If cracks have appeared, has the frequency of their occurrence diminished when compared to the cracks which have appeared in boilers of the same type as mentioned in the paper but in which the tubes were rolled-in by methods other than the one described by the authors?

(d) Have the authors observed circular cracks in the ends of the rolled-in tubes, and if so, have they ascertained the reason for these cracks?

(e) Do the authors know of an indicator suitable for determining the tube flare within the shell?

(f) In what units is the thickness of the tube wall given? The writer has inferred that the "gage" referred to by the authors means American standard plate gage.

Authors' Closure

The authors keenly appreciate the interest taken in their efforts to bring the matter of boiler-tube expansion to the attention of engineers. They are also not unmindful of the fact that they have only made a beginning of studying this subject and that many questions remain to be answered. The questions raised in the discussion are such that each discussion will be given a separate reply.

Mr. Skog raises the patent question of cost of application of the elongation method. In the application of this method no more help is used and no more time is required than in the application of the older methods. When the cost of rerolling tubes, improperly rolled initially, is given its proper weight the elongation method is actually less expensive than the older methods. It is not necessary to complete one row of tubes at a time, as suggested. No material change has been made in method of assembly usually used.

The tubes were bought on Specification A.S.T.M. Designation A83-33. Table III of this specification gives the following tolerances for tubes 4 in. in outside diameter and smaller:

Outside diameter, over $^1/_{64}$ in., under $^1/_{32}$ in.

Wall thickness, over 3 Bwg (Birmingham wire gage) numbers. This thickness tolerance applies only to eccentricity so that the average variation would be about $1^1/_2$ Bwg numbers.

Tube-hole diameter tolerance is specified in the A.S.M.E. Boiler Construction Code, Combined Edition (1931 with all errata and changes inserted), Section 6, I-19 on Rules for Inspection. The maximum size of tube hole is specified not to exceed *nominal* plus $^1/_{32}$ in. The implied meaning is obvious. A $3^1/_4$-in. tube may be anything from 3.219 in. to 3.266 in. in outside diameter and the tube hole may be anything from 3.250 in. to 3.281 in. in diameter. The tube-wall thickness may vary according to the schedule given in Table 1 of this discussion.

TABLE 1

Nominal thickness, Bwg	Min, in.	Max, in.	Tolerance, in.
2	0.284	0.320	0.036
4	0.238	0.271	0.033
6	0.203	0.229	0.026
8	0.165	0.191	0.026
10	0.134	0.155	0.021

Table 1 gives only a partial list but illustrates the implication of the specification. The authors' experience with tubes and tube holes indicates that the A.S.M.E. Boiler Construction Code has been met.

Mr. Skog's comment regarding the indication of contact in the case of a tube of noncircular section is correct. This condition presents no difficulties because no elongation will occur until full contact has been established and in consequence there will be no movement of the indicator needle until full contact. There is a second condition which Mr. Skog has not noted which merits comment. The tube may not be coaxial with the hole during the first phase of the expanding operation. Contact occurs and further rolling may produce a positive or negative indication as the tube is being forced into the correct position and full contact. At this instant the needle will pause and elongation will be measured from this point. The application of the elongation method to

such exceptional and abnormal cases can be easily worked out by the alert tube installer.

Experience has not shown the necessity for predetermining exactly the proper elongation for different combinations of tube diameter and sheet thickness. All tests made by the authors have shown that the relation between holding strength and elongation becomes a maximum at about the same elongation, namely, 0.020 in. for tubes of different sizes rolled into sheets of different thicknesses, so that meticulous care is not called for in the actual assembly of the joints by this method.

In their investigations into expanded-tube joints the authors have not been satisfied only with securing a simple method for rolling-in tubes, so that the erecting job may be carried out with the least trouble, but have constantly maintained an interest in the whole life of the tube. It has not been taken for granted that a certain portion of all tube joints must fail, but rather that a failed joint shows either that the tube was improperly made or that the joint was not properly expanded. The meeting of a construction code has not been made the criterion of quality in rolled boiler tube joints. In the past when pressures were lower than those now common and when boilers were comparatively simple structures without waterwalls and other modern developments, it was not necessary for the builder to be concerned about the factor of safety of tube joints. Any rolling-in method would work because the strength of the joint was many times the boiler test pressure. No one thought much about factor of safety of boiler tube joints. Only within the past year a case has come to the authors' attention in which the holding strength of the joint was less than the code test pressure. The factor of safety of the joints, based on boiler code test pressure, was less than one. It now becomes necessary to seriously consider developing the maximum holding strength of which the tube and tube sheet is capable and this demands definite control and uniform practice in the making of tube joints.

Mr. O'Brien has raised an interesting question in relation to the production of a fixed elongation in the last tube expanded in any tube bundle. The authors have no data on which to base a quantitative answer but the following phenomena have been observed: In the case of the long straight tube the tube reacts as a long column having fixed ends and consequently deflects from a straight line. This same phenomenon is observed when tubes of a given bundle are expanded unequal amounts. The bent tube, of course, is much more flexible than the straight one and consequently a greater portion of the axial force resulting from expansion is dissipated in change of shape than is true in the case of the straight tube. The short tube presents a more difficult problem but a tube so short that it could not be easily expanded to the proper elongation has not come to the authors' notice. The notes in the paper relating to the axial force set up in the tube during expansion give some idea of the change in axial force with change of measured elongation.

Mr. Fellows has touched on a phase of this question which the authors believe has not been given the consideration that it deserves. Tubes have been rolled into drums without enough regard for the deformation of ligaments or the disturbance of metal in the tube wall. Points of very high stress have been set up due to the rolling-in operation and pitting and ultimate failure has been the result. In the May, 1935, issue of *Mechanical Engineering* there was published an article entitled, "Boiler Steel Embrittlement," by E. P. Partridge and W. C. Schroeder, both of U. S. Bureau of Mines, Rutgers University, New Brunswick, N. J. The first two generalizations presented by these authors are quoted as follows:

"(1) The cracking of steel subject to chemical attack while under stress consistently commences in a region where the local stress is greater than the average.

(2) An unknown but probably very high local stress is necessary to initial cracking."

One case which the authors have studied will be discussed in order to emphasize the importance of the application of the previously mentioned conclusions to the rolled-in tube joints. At the Trenton Channel Power Plant there were many failures of "tack" tubes. These tubes all failed in the same manner. Figs. 3 and 4 of this discussion illustrate the location and manner of failure. The walls of the tubes were eaten through circumferentially about $^3/_8$ in. before the inner end of the rolled section. These tubes had been expanded using a four-roll expander in

Fig. 3 Tube Wall Corroded Through at the Point of Greatest Disturbance of the Metal in the Wall

Fig. 4 Tubes Showing Well-Developed Corrosion at Point of Highest Stress

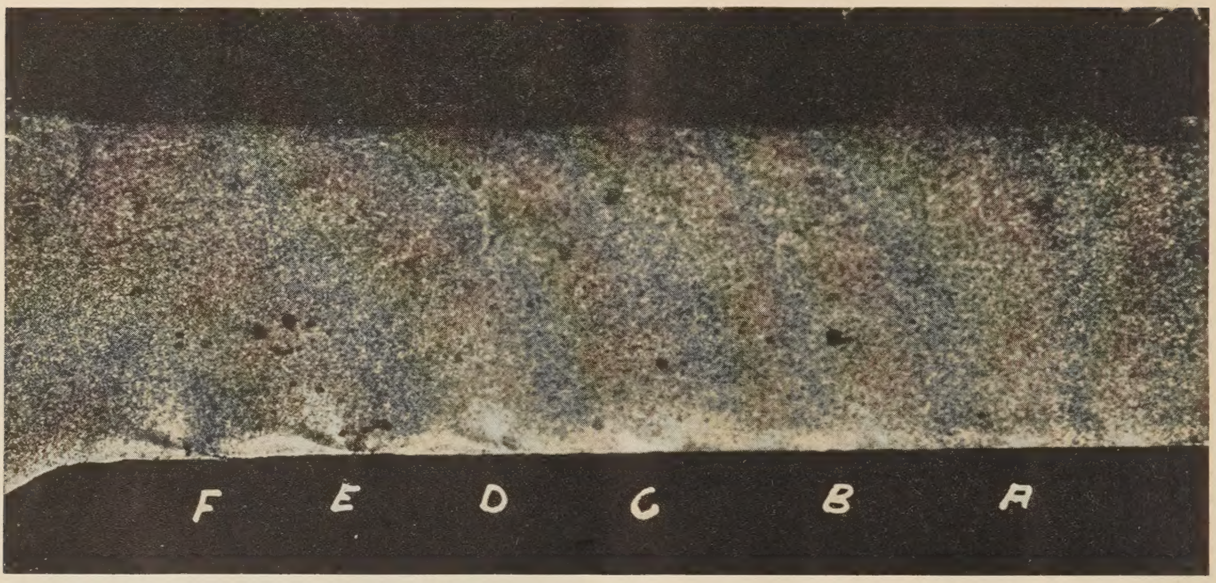

Fig. 5 Section of Rolled Tube Showing Points of Greatest Disturbance of the Metal (×8)

which the rolls had short-radius corners on the entrance ends. Reduction in wall thickness was not excessive, being of the order of 7 to 9 per cent. The replacing tubes were expanded using the same type of tool but having the entrance ends of the rolls ground to a long radius of 5 in. These replacing tubes have been in service about four years and have shown no evidence of a repetition of the corrosion condition.

Several theories were advanced to explain the condition. Finally the authors undertook to repeat the condition set up in the tubes which failed. A tube of about the same physical characteristics as the failed tubes was rolled into a tube hole using a tool equipped with the same shape of rolls as were used originally on the tubes which failed. The reduction in wall thickness, while somewhat greater than that found in the failed tubes, was not excessive. Fig. 5 of this discussion is a photomicrograph (×8) of a polished etched section through the rolled portion of tube wall. The etching fluid was 10 per cent HNO_3. It is to be noted that there are points of high stress at *A*, *B*, *C*, *D*, *E*, and *F* which correspond to the spiral formed by the entrance ends of the rolls as they advanced into the tube. The condition at *F* calls for special notice as it is exactly the same distance from the end of the rolled section as was the corrosion in the tubes shown in Figs. 3 and 4, namely, about $^3/_8$ in. A photomicrograph (×100) was taken in region *F*. This is shown in Fig. 6 of this discussion. An examination of Figs. 5 and 6 shows the cause of the corrosion to be none other than excessive cold working of the metal at region *F*. A very high stress was locked up in the wall of the tube in this region. A cul-de-sac had been formed by the rolling operation. The necessary conditions for localized corrosion were present and only about two

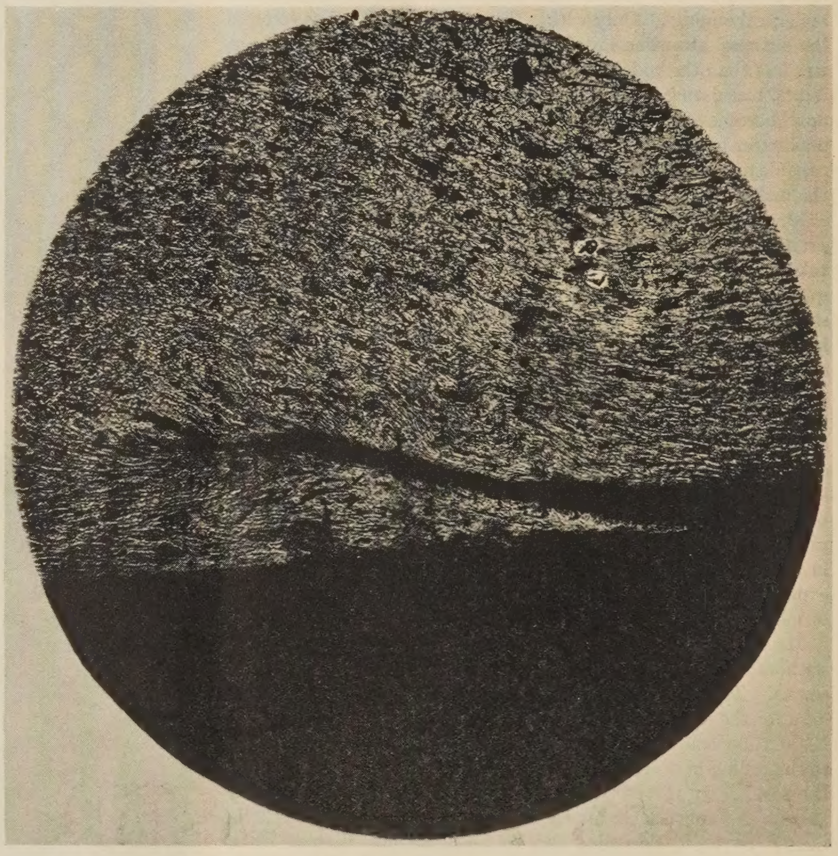

Fig. 6 Section of Fold Metal on Inside of Tube (×100)

years of service saw the end of these tubes. Many of these tubes have failed and the cost of replacement has been charged to maintenance. A change in roller shape appears to have corrected these conditions. The new rollers set up much less local stress because the surface of the tube is not disturbed so violently and no cul-de-sac has been found. The authors believe, but have no experimental proofs, that ligament failures in many cases can be charged to excessive plastic deformation due to excessive rolling. This phase of the problem of producing permanent rolled joints is one which deserves serious study.

The authors are pleased to have a discussion of their paper by an engineer having European point of view and background. It is probably not generally known among American mechanical engineers that the question of boiler-tube joints has been made the subject of considerable study in Europe for the past ten years. There are many publications telling of these researches. One of the authors of this paper under discussion has recently completed a review of four of these papers, three of which were published in Germany and one in Switzerland. Based on the cost of similar research carried on by the authors, the work represented by these four papers must have cost over $50,000. One of the papers speaks of tests on 4975 experimental rolled joints. Mr. Noeff's discussion is therefore most welcome as it gives an opportunity to refer to these European researches.

Mr. Noeff is correct in stating that the authors have not shown the relationship between elongation and "(*a*) thickness of the tube wall, (*b*) the decrease in the thickness of the tube wall after it has been rolled, (*c*) the width of the inrolled ring of the tube in the tube sheet." These dimensions were all carefully measured and recorded but were not evaluated against the elongation. After several unsuccessful attempts to account for the metal in the rolled-in section of the tube by attempting to apply to actual rolled joints the same formula as Mr. Noeff develops in his Equation [1], which is of course approximately correct, they centered their attention on the major purpose of tube rolling. The purpose of rolling-in a tube into a tube hole is to produce the strongest possible joint and one which will not only meet the pressure test specified by the code, but will continue to perform satisfactorily throughout the anticipated life of the boiler. This implies the rolling of tube joints so as to produce the least possible cold-working of the tube wall and the tube-sheet metal. The meeting of these requirements called for a check of the change in tube-wall thickness and in tube-hole diameter resulting from the expanding operation, in consequence these measurements were made.

The research reported on in the Thum and Jantscha paper which Mr. Noeff refers to were carried out on tubes machined inside and out to a close micrometer tolerance and the tube holes were in no case more than 0.2 mm (0.008 in.) larger than the outside diameter of the tube. The results of tests carried out on such ideal parts can hardly be applied to the assembling of tubes and sheets such as are found in American practice. The Code and A.S.T.M. specification tolerances are noted in the answer to Mr. Skog's discussion. The authors have used only parts having such variations from the ideal as are found in an actual boiler. If commercial manufacturers could furnish tubes and tube holes having dimensions to as close micrometer tolerance as those used by Thum and Jantscha in the studies referred to previously at a price which the purchaser could afford to pay and if the design of the boiler were such that parts made to such close tolerance could be assembled by the available erectors, the mathematical method of Thum and Jantscha would apply. But when the erecting engineer has to get along with such variations in tube diameter and wall thickness and tube-hole diameter as are met with in American commercial practice, these formulas cannot be applied if one hopes to produce joints of anywhere near uniform strength. If he further hopes to not overroll his tubes or tube sheets he must find a more flexible method of control than that afforded by the uniform-entrance method.

The authors do not care to enter into a discussion of the work of Thum and Jantscha at this time. However, their conclusion after a careful study of the work of these investigators and of others published in German is that the amount of tube expansion which they advocate is at least twice as much as is normally produced by the application of 0.020 in. to 0.025 in. elongation as used in boiler erection in the authors' company. In some instances the expansion recommended was so great that its application would result in a reduction of tube-wall thickness of as much as 25 per cent with attendant work hardening and "tendency to form cracks."

The authors do not agree with Mr. Noeff that with the materials usually found in modern boiler practice, the amount of expansion is a function of the tensile strength, ductility, yield point, etc., of the tubes and sheets. If Mr. Noeff refers to strength of the rolled-in joint the authors are willing to go much further than he. Tests conducted and not yet reported on show that the holding strengths of rolled-in joints vary with (1) condition of finish of tube and tube hole, (2) tube-sheet thickness, (3) tube diameter, (4) tube-wall thickness, (5) degree of expansion, and (6) difference in hardness of sheet and tube.

In no case are all the tubes of a boiler installed at one time. While this might be possible in a small boiler, in large boilers of the vertical-tube type so common in America, such procedure would make it impossible to adjust each tube to its proper axial position before rolling-in the joints. Usually not more than six rows of tubes are inserted at once. This number depends on what appears to be the easiest way to handle the particular tube arrangement for convenience in final adjustment. With six rows of tubes installed at once there is no difficulty in using the dial-indicator method described.

The problem of accumulated elongation is perhaps more complex than is suggested in the paper. Its existence is best illustrated by a concrete example. In a six-drum Stirling-type boiler of about 30,000 sq ft heating surface the two top drums on each side were connected by 35 bent circulating tubes 4-in. in diameter. By mistake these tubes on one side of the first boiler erected were rolled-in beginning at one end and proceeding in sequence to the other end. The accumulated elongation in this case was such that the lighter drum was lifted $^1/_{16}$ in. off the supporting saddle on the end at which the rolling job was completed. The drum lifted weighed 35,000 lb and carried about 10,000 lb of tubes already attached to it. On the second side of this same boiler the rolling-in was done in four parts. The job was started at the middle of the length of the drum and proceeded one quarter of the length of the drum first on the right then on the left of the starting point. The tubes in the remaining two quarters of the length of the drum were rolled-in starting at the ends of the drum and proceeding toward the middle. There was no measurable lifting of the lighter drum in this case. Similar portions of the three other boilers of the same design have been treated in the same manner and with the same result. Fig. 7 of this discussion is introduced to illustrate the effect of sequence of rolling-in tubes. The scale is of course greatly exaggerated.

The answers to Mr. Noeff's listed questions follow:

(*a*) The use of the elongation method of rolling-in boiler tubes has resulted in a great reduction in the number of rerolled tubes necessary to pass inspection. After five years service in the case of one boiler there has been no report of tube end failure. In the authors' company, the tubes of five new boilers and the tubes of three rebuilt waterwalls have been installed using the elongation method. In all there are probably 20,000 tube joints and no failures have been reported. Some of these joints are operating

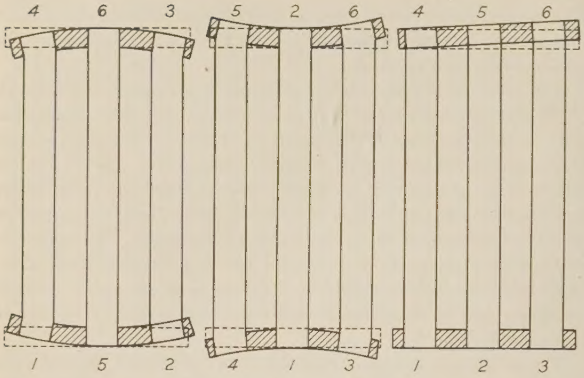

Fig. 7 Effect of Sequence of Rolling

at 425 lb per sq in. gage and the remainder at 700 lb per sq in. gage. There are also many boilers in which this method was used in plants outside the authors' company, other than those at the Ford Motor Company and the Springwells Plant of the Detroit Water Commission on which no report is available.

(*b*) and (*c*) In the authors' company there is no knowledge of cracks in the rolled-in portions of boiler tubes except those resulting from corrosion fatigue. This form of failure is illustrated in Figs. 4 and 5 of this discussion and its probable cause in Figs. 6 and 7. The description of these figures and the method of correcting the fault is described in the accompanying text.

(*d*) The authors do not understand exactly what form of failure the writer refers to but suspect he means circumferential cracks not associated with corrosion; such forms of failure as are illustrated in Fig. 8 of the discussion. This figure was taken from the paper, "Über das Einwalzen von Rohren unter besunderer Berucksichtigungder Frage der Rundvisse in den Einwalzstellen von Siederohren," by A. Thum and W. Ruttman, published in Mitteilungen Nr. 45 der Vereinigung der Grossekesselbesitzer.

With only these illustrations available to judge from, it is impossible to form a definite conclusion. It is the authors' opinion, however, based on their experience with rolling-in tubes, that these cracks are the result of high stress concentration resulting from either greatly exaggerated rolling, or the use of improperly shaped or defective tools or a combination of the two. A case of cracked brass tubes has come to the authors' attention. These tubes were in an air cooler on a 10,000-kw turbogenerator. The tubes failed at the rolled-in section after six months' service. An examination of these tubes showed that some failed in tension and some in compression. It appeared that the cause of the failures was due to excessive rolling and improper sequence of rolling. The replacing tubes were rolled to about 0.020-in. elongation and with due care as to sequence so that the accumulated elongation was not sufficient to seriously distort the tube sheet. These replacing tubes have been in almost continuous service for three years.

(*e*) The authors have had no experience with any kind of indicating device for controlling flare rolling.

(*f*) The ordinary method of designating tube-wall thickness used in America is the Birmingham wire gage (Bwg).

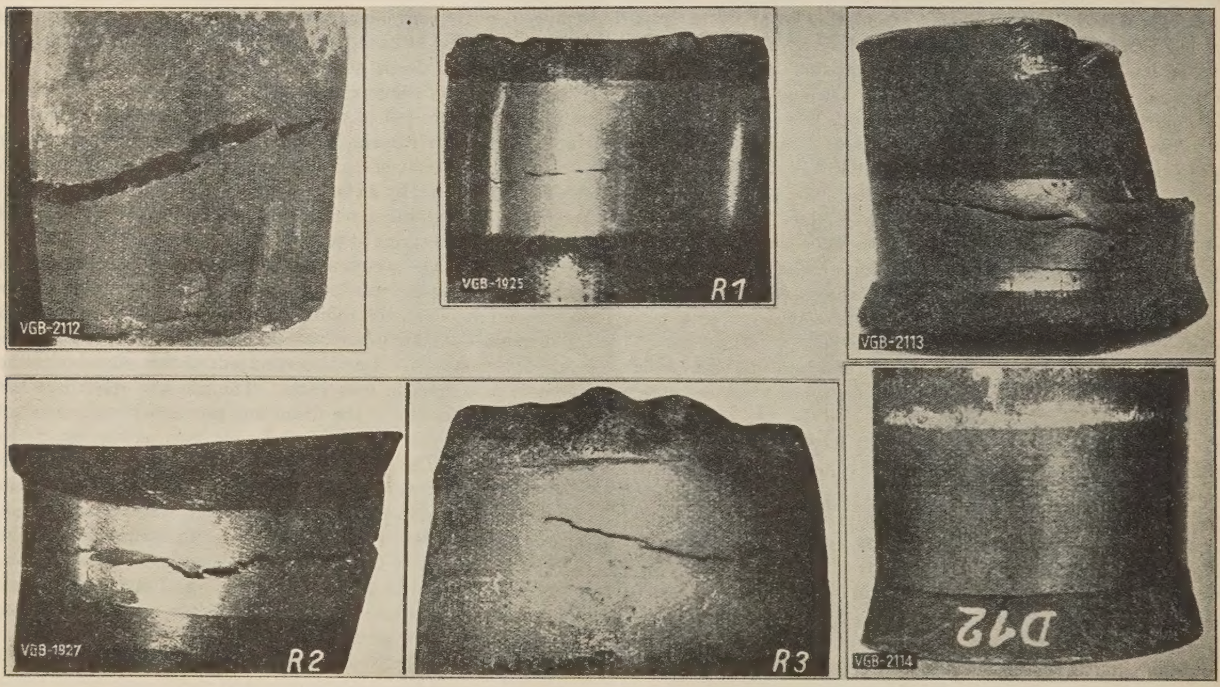

Fig. 8 Tube Ends With Circumferential Cracks